THE ELEMENTS

OF

MECHANICAL ENGINEERING

PREPARED FOR STUDENTS OF

THE INTERNATIONAL CORRESPONDENCE SCHOOLS

SCRANTON, PA.

Volume II

STEAM AND STEAM ENGINES

STRENGTH OF MATERIALS

APPLIED MECHANICS

STEAM BOILERS

WITH PRACTICAL QUESTIONS AND EXAMPLES

First Edition

SCRANTON

THE COLLIERY ENGINEER CO.

1897

BURR PRINTING HOUSE,
FRANKFORT AND JACOB STREETS,
NEW YORK.

CONTENTS.

STEAM AND STEAM ENGINES.

STEAM.

1189. Steam is water vapor—that is, it is water changed to a gaseous state by the application of heat.

1190. The effects of the application of heat to water may be shown by the following simple experiment: Take a cylinder of indefinite length, fitted with a piston, which, let us suppose, has no weight, and place within the cylinder say 100 cubic inches of water. For convenience, let the area of the cross-section of the cylinder be 100 square inches; then, the height of the water in the cylinder is 1 inch. (See (*a*), Fig. 235.) Assume the temperature of the water to be 32° F.—that is, just at freezing point. The upper end of the cylinder is open to the atmosphere, and the piston and the water under it are, therefore, subjected to a pressure of 14.69 pounds per square inch, or a total pressure of $100 \times 14.69 = 1{,}469$ pounds. Now, let heat be applied to the water in the cylinder; what will take place?

The temperature will gradually rise from 32°. The volume of the water will decrease slightly, until the temperature reaches 39.2° F., and then will increase gradually, though almost imperceptibly, until the temperature reaches 212° F. At that point the piston will be 1.043 inches from the bottom of the cylinder, instead of 1 inch, the volume of the water thus having increased 4.3%, while the temperature has risen from 32° F. to 212° F. Just at this point the vibrations of the molecules of the water, which have become more and more rapid as the temperature has risen, have become rapid enough not only to overcome the force of cohesion, which tends to keep them together, but also to raise the piston with its load of 1,469 pounds. The water

begins to change to the gaseous form, or, in other words, begins to become steam. The temperature now remains at 212°, and can not be raised above that point as long as any water remains in the bottom of the cylinder.

1191. If, after the water is heated to 212°, more heat is added and the pressure remains at 14.69 pounds per square inch, there will be no rise in temperature, but some of the water will change to steam and the piston will rise as shown at (*b*), Fig. 235. Now, push the piston down until the volume of the steam is reduced one-half, withdrawing heat so as to keep the temperature at 212°. We know that, with a perfect gas, this decrease in volume would double the pressure; with steam, however, the only effect is to change part of the steam to water without changing the pressure.

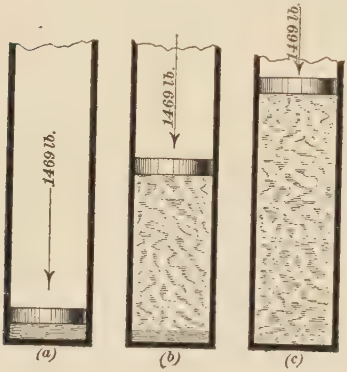


FIG. 235.

Now, raise the piston, keeping the temperature at 212°. With a perfect gas, we know that the pressure would fall in exact proportion to the increase in volume; the steam, however, behaves in a different manner. Part of the water changes to steam as the piston rises, but there is no change in the pressure.

If the pressure on the piston is increased, it is found that the temperature of the water must be increased before steam is formed; thus, make the pressure 30 pounds per square inch, and we find that no steam is formed until the temperature reaches 250.3°. At this temperature, it is also found that no change in the volume of the steam affects its pressure.

We have learned that the pressure of a perfect gas depends on its temperature and volume; the above experiments, however, show that the pressure of steam in contact with

water is independent of the volume, but depends entirely on the temperature; if there is no change in temperature, the only effect of a change in volume is to change the quantity of steam.

Steam in contact with water is called **saturated steam.** If not in contact with water, steam is still saturated if its temperature is the same as that at which water boils when subjected to the same pressure.

The conditions of saturation apply as well to all other vapors as to steam; any vapor in contact with the liquid from which it is formed is *saturated*, and for a given temperature, can have but one pressure.

1192. Suppose, now, that just enough heat is applied to change all the water into steam, as in (*c*), Fig. 235. Up to the point where the last drop of water is evaporated, the steam is *saturated*, and at the pressure of 14.69 pounds per square inch, the temperature remains at 212°. After the last drop of water is evaporated, there is a change in the effect produced when heat is applied. Both the temperature and volume of the steam now increase, and in many respects its properties resemble those of a perfect gas. Steam in this condition is said to be **superheated.** The specific heat of superheated steam under constant pressure is .48.

1193. To raise each pound of the water from 32° to 212° required the expenditure of 180.531 B. T. U. If the specific heat of water were 1 at all temperatures, the heat required would have been $212 - 32 = 180$ B. T. U., but at high temperatures the specific heat of water is a little greater than 1, consequently a little more than 180 B. T. U. are required.

The work expended in raising 1 pound of water from 32° to 212° may be found by multiplying the number of heat units required by 778, the mechanical equivalent of heat; it is therefore $180.531 \times 778 = 140{,}453.12$ foot-pounds.

To change each pound of water at 212° to steam at 212°

requires the expenditure of 966.069 B. T. U., or 966.069 × 778 = 751,601.68 foot-pounds of work.

Part of this work was spent in lifting the piston against the atmospheric pressure, and part in overcoming molecular forces. The former is called **outer** or **external work**; the latter, **inner** or **internal work.**

The outer work may be calculated as follows: At the beginning of the operation the piston was 1 inch from the bottom of the cylinder; when all the water had been changed to steam, the piston was 1,646 inches above the bottom. It had therefore been raised 1,646 − 1 = 1,645 inches = $137\frac{1}{12}$ feet, against a constant pressure of 1,469 pounds, and the work done was 1,469 × $137\frac{1}{12}$ = 201,375.417 foot-pounds, of which all but the very small portion represented by the expansion of the water when heated from 32° to 212° was developed during the formation of the steam. At a temperature of 32°, 100 cubic inches of water weigh 3.6121 pounds; hence, the external work per pound of water was 201,375.417 ÷ 3.6121 = 55,750.2 foot-pounds. The total work of changing 1 pound of water to steam at 212° was 751,601.68 foot-pounds; of this, nearly 55,750.2 foot-pounds were spent in external work, leaving 751,601.68 − 55,750.2 = 695,851.48 foot-pounds as the internal work.

1194. Let the experiment be tried again with tne same amount of water in the cylinder, but with a weight on the piston in addition to the atmospheric pressure. Let the pressure be say 32 pounds per square inch, the total pressure being 32 × 100 = 3,200 pounds. It will now be found that steam does not begin to form until the temperature has reached 254° F. This is easily accounted for, since the vibrations of the molecules of the water would necessarily have to be more rapid, that is, the temperature would have to be higher, in order to overcome the increased pressure on the piston.

It will now require 223.021 B. T. U. = 173,510.34 foot-pounds to raise 1 pound of the water from 32° to 254°. It will require 936.389 B. T. U. = 728.510,64 foot-pounds to

change one pound of the water at 254° into steam of the same temperature. When the water is all evaporated, the piston will be found at a height of 791.8 inches from the bottom of the cylinder. It has been raised by the steam to a height of $791.8 - 1 = 790.8$ inches $= 65.9$ feet, against a constant pressure of 3,200 pounds. The total outer work is, therefore, $65.9 \times 3,200 = 210,880$ foot-pounds. The outer work per pound of water is $\frac{210,880}{3.6121} = 58,381.55$ foot-pounds. The inner work is $728,510.64 - 58,381.55 = 670,129.09$ foot-pounds.

1195. It is thus apparent that changing the pressure on the piston has changed all the other conditions—the temperature, the volume, the inner and outer works, and the heat required to raise the water from 32° to the boiling point.

The relations between these various properties of steam can not be expressed by any general law. They have been determined experimentally, however, and, taken together, form what are known as the steam tables.

STEAM TABLES.

RELATION BETWEEN PRESSURE AND TEMPERATURE OF SATURATED STEAM.

1196. It has been shown that the temperature of saturated steam depends *only* upon the pressure, and not at all upon the volume or any other property of the steam; conversely, the pressure depends upon the temperature.

If the steam in a boiler shows a certain pressure by the gauge, we know that it can only have one temperature. In fact, it would be possible, though, perhaps, not convenient, to determine the pressure of steam from the reading of a thermometer inserted into it. At the atmospheric pressure of 14.69 pounds per square inch, saturated steam *always* has a temperature of 212° F.; at a pressure of 79 pounds per square inch above vacuum, steam can have only a temperature of 311° F.

1197. The temperatures of saturated steam corresponding to the different pressures have been experimentally determined with great care and accuracy by the French physicist, Regnault, and are given in the Table of the Properties of Saturated Steam. Column 1 of the steam table contains the pressures from 1 to 300 pounds per square inch, and column 2 contains the corresponding temperatures.

A large number of formulas have been devised to express the relation between the pressure and temperature of saturated steam. The following by Rankine is probably as accurate and convenient as any:

Let t = temperature, Fahrenheit;

T = absolute temperature $= t + 460°$;

p = absolute pressure (pressure above vacuum) in pounds per square inch.

$$\text{Then, } \log p = 6.1007 - \frac{2,719.78}{T} - \frac{400,215}{T^2}. \qquad \textbf{(90.)}$$

EXAMPLE.—Find the pressure corresponding to 285.8° *F.*

SOLUTION.—$T = 285.8 + 460 = 745.8°$.

$$\text{Log } p = 6.1007 - \frac{2,719.78}{745.8} - \frac{400,215}{745.8^2} = 1.73437.$$

The number corresponding to this logarithm is 54.246. Hence, the pressure is 54.246 lb. per sq. in. The table gives 54 lb.

EXAMPLES FOR PRACTICE.

1. The temperature of steam being 224°, what is the pressure?

Ans. 18.557 lb. per sq. in.

2. Find the pressure in the last example by using formula **90.**

Ans. 18.574 lb. per sq. in.

3. Find the pressure corresponding to a temperature of 312°, (*a*) by means of the steam tables; (*b*) by formula **90.**

Ans. $\begin{cases} (a) \text{ 80.157 lb. per sq. in.} \\ (b) \text{ 80.566 lb. per sq. in.} \end{cases}$

4. The pressure being 105 pounds absolute, (*a*) find the temperature by formula **90**; (*b*) what is the difference between the above result and the value given in the tables?

Ans. $\begin{cases} (a) \text{ 330.756°.} \\ (b) \text{ .413°.} \end{cases}$

5. Assuming that there has been no heat added or taken away, what is the difference in pressure due to a fall of temperature from 281.7° to 259°? Ans. 15.764 lb. per sq. in.

NOTE.—All examples under the heading *Examples for Practice* should be worked out by the student. The student should solve them in all cases before proceeding further.

1198. Heat of the Liquid.—Column 3 of the table gives the heat required to raise *one pound* of water from 32° F. to the temperature given in column 2. It will be observed that it is nearly the same as the difference between the given temperature and 32°. For instance, the heat of the liquid corresponding to a pressure of 60 pounds, and a temperature of 292.575°, is 262.248 B. T. U. The difference in temperatures is $292.575° - 32° = 260.575°$. In many steam tables this column is left out, and the heat of the liquid is found for any temperature by subtracting 32° from the temperature. This method is usually exact enough for practical purposes. The heat required to raise the temperature of a pound of water from 32° to the boiling point is called the **heat of the liquid.**

The results of column 3 have been determined experimentally.

1199. The **total heat of vaporization** for any given temperature is the number of heat units required to raise a pound of water from 32° F. to the given temperature, and change it into steam at that temperature. In the first experiment, shown in Fig. 235, the total heat of vaporization was the total heat applied to the cylinder from the beginning of the experiment until the water was all turned into steam. It is the sum of the heat of the liquid, the internal work, and the external work.

Referring back to the first experiment, the heat of the liquid was found to be 180.531 B. T. U. per pound, or, expressed in units of work, 140,453.12 foot-pounds. The heat used to perform outer work was found to be equivalent to 55,750.2 foot-pounds, and that used in performing inner work was equivalent to 695,851.48 foot-pounds. The

total heat of vaporization, then, is equivalent to the sum of all these works, or to 140,453.12 + 55,750.02 + 695,851.48 = 892,054.8 foot-pounds, or, expressed in thermal units, $\frac{892,054.8}{778} = 1,146.6$ B. T. U., the same as given in the steam table.

In the second experiment, the work equivalent to the heat of the liquid was 173,510.34 foot-pounds, the outer work 58,381.55 foot-pounds, and the inner work 670,129.09 foot-pounds. The total heat of vaporization (expressed in foot-pounds) is, therefore,

58,381.55 + 173,510.34 + 670,129.09 = 902,020.98 foot-pounds, or $\frac{902,020.98}{778} = 1,159.41$ B. T. U. per pound.

1200. The total heat of vaporization has also been experimentally determined by Regnault for all pressures and temperatures. The following formula expresses the results of the experiments very closely:

$$H = 1,081.94 + .305\ t, \qquad \textbf{(91.)}$$

in which H is the total heat of vaporization, expressed in heat units, and t the temperature in degrees F.

EXAMPLE.—What is the total heat of a pound of steam when the absolute pressure is 50 pounds per square inch?

SOLUTION.—Looking in column 2, the temperature corresponding to a pressure of 50 pounds is 280.904° F. Substituting this in formula **91**,

$$H = 1,081.94 + .305 \times 280.904° = 1,167.615 \text{ B. T. U. Ans.}$$

The **total heat of vaporization** is given in column 5 of the steam table.

1201. Latent Heat of Vaporization.—In the first experiment it was found that a certain number of B. T. U.'s were required to raise a pound of water from 32° to the temperature of the boiling point, and that another quantity of heat was required to change the water at boiling point into steam at the same temperature. The former of these quantities of heat we called the heat of the liquid; the latter is the **latent heat of vaporization,** or simply the **latent heat of steam.** The heat required for both of the above

operations was defined as the **total heat.** Therefore, letting H represent the total heat, q the heat of the liquid, and r the latent heat of vaporization, we have the following equation:

$$H = q + r.$$

$$\text{Therefore, } r = H - q.$$

Hence, to find the latent heat of vaporization, subtract the heat of the liquid from the total heat. The latent heat is given in column 4 of the table, and is obtained by simply subtracting the numbers in column 3 from those in column 5.

EXAMPLES FOR PRACTICE.

1. What is the total heat of 25 pounds of saturated steam having a temperature of 340°? Ans. 29,640.975 B. T. U.

2. What is the total heat of a pound of steam at a pressure of 90 pounds gauge? Ans. 1,182.88 B. T. U.

3. Solve both of the above problems by means of formula **91.**

Ans. $\begin{cases} 29,641 \text{ B. T. U.} \\ 1,182.882 \text{ B. T. U.} \end{cases}$

4. What is the latent heat of vaporization corresponding to a pressure of 85 pounds absolute? Ans. 892.086 B. T. U.

5. Find the heat of the liquid of 10 pounds of water at a temperature of 297.6°. Ans. 2,673.77 B. T. U.

6. What is the latent heat of vaporization of the water in example 5? Ans. 9,053.31 B. T. U.

1202. The **specific volume** of saturated steam has been determined both experimentally and by complicated mathematical formulas. Column 7 of the table gives the volume in cubic feet of a pound of steam at the different pressures, and column 8 the ratio between the volume of a pound of steam at the temperature in column 2 and a pound of water at maximum density.

1203. The results in column 7 may be found approximately by Rankine's formula:

$$p\,V^{\frac{17}{16}} = 475, \qquad \textbf{(92.)}$$

in which p is the pressure in pounds per square inch, and V is the volume in cubic feet of a pound of steam at the given pressure.

EXAMPLE.—Find the volume of one pound of steam at a pressure of 20 pounds per square inch absolute.

SOLUTION.—
$$20\, V^{\frac{17}{16}} = 475$$
$$V^{\frac{17}{16}} = \frac{475}{20} = 23.75$$
$$\log V^{\frac{17}{16}} = \tfrac{17}{16} \log V = \log 23.75 = 1.37566$$
$$\log V = 1.37566 \times \tfrac{16}{17} = 1.29474$$
$V = 19.71$ cubic feet. The table gives 19.73 cubic feet.

Column 8 can be obtained from column 7 by multiplying by 62.425, the weight in pounds of a cubic foot of water.

EXAMPLES FOR PRACTICE.

1. Find, by means of the tables, the volume of 12 pounds of saturated steam having a gauge pressure of 101 pounds. Ans. 45.42 cu. ft.

2. Solve the above example by formula **92.** Ans. 45.336 cu. ft.

3. The volume of 5 pounds of saturated steam is 22 cubic feet; (*a*) find the pressure by formula **92**; (*b*) find the weight corresponding to this pressure from the tables. Ans. { (*a*) 98.405 lb. per sq. in. (*b*) .226829 lb. per cu. ft. }

4. If 2.87 cubic feet of steam expand until the volume and pressure are, respectively, 9 cubic feet and $2\frac{1}{2}$ pounds gauge, what was the original pressure? Ans. 57.93 lb. per sq. in.

5. The steam in an engine cylinder has a pressure of 85 pounds gauge, and a volume of 1.6 cubic feet at cut-off. It expands to a pressure of 18 pounds absolute; what is the final volume? Use formula **92,** in both this and the last example, remembering that if $p\, V^{\frac{17}{16}} = 475$, $p_1\, V_1^{\frac{17}{16}} = 475$ also, and, consequently, that $p\, V^{\frac{17}{16}} = p_1\, V_1^{\frac{17}{16}}$. Ans. 8.0134 cu. ft.

1204. The **density of saturated steam** (that is, the weight in pounds of a cubic foot for the different pressures and temperatures) is given in column 6.

The density is the reciprocal of the specific volume. For example, a pound of steam at 44 pounds pressure occupies a volume of 9.403 cu. ft.; therefore, one cubic foot of steam

at 44 pounds pressure must weigh $\frac{1}{9.403} = .1063$ pound. Hence, the values in column 6 may be obtained by taking the reciprocals of those in column 7.

1205. The above properties of saturated steam compose the Table of the Properties of Saturated Steam. Two other properties are given in some steam tables; viz., the heat corresponding to the inner work, and the heat corresponding to the outer work, the sum of the two being the latent heat of vaporization. As these properties are not of much practical importance, they have been omitted from the table.

It must always be borne in mind that, in using this table, the pressures are reckoned from vacuum, and *not* from atmospheric pressure. That is, 14.7 pounds must be added to all gauge pressures to make them available for use in the steam table. Again, the heat of the liquid, the total heat, and the latent heat are calculated from 32° F., *not* 0° F. A great deal of trouble will be saved by carefully remembering these points.

EXAMPLES ON THE USE OF THE STEAM TABLE.

1206. Example 1.—Find the heat required to change 6 pounds of water at 32° into steam at 84 pounds pressure absolute.

Solution.—The total heat of one pound at 84 pounds pressure is, from column 5, 1,178.091 B. T. U.

$1,178.091 \times 6 = 7,068.546$ B. T. U. Ans.

Example 2.—How many heat units are required to raise $8\frac{1}{2}$ pounds of water from 32° to 250° F.?

Solution.—Looking in column 3, the heat of the liquid of one pound at 250.293° is 219.261 B. T. U. $219.261 - .293 = 218.968$ B. T. U. $= q$ for 250°.

Then, for $8\frac{1}{2}$ pounds, it is $218.968 \times 8\frac{1}{2} = 1,861.228$ B. T. U. Ans.

Example 3.—How many foot-pounds of work will it require to change 60 pounds of water at 311° F. into steam at the same temperature?

SOLUTION.—Looking under column 4, the latent heat of vaporization is 895.729; that is, it takes 895.729 B. T. U. to change one pound of water at 311° into steam at the same temperature. Therefore, it takes 895.729 × 60 = 53,743.74 B. T. U. to perform the same operation upon 60 pounds of water.

53,743.74 B. T. U. = 41,812,629.72 foot-pounds. Ans.

EXAMPLE 4.—Find the volume occupied by 14 pounds of steam at 30 pounds *gauge* pressure.

SOLUTION.—30 pounds gauge pressure = 30 + 14.7 = 44.7 pounds absolute pressure. The nearest pressure in the table is 44 pounds, and the volume of a pound at that pressure is 9.403 cu. ft. The volume of a pound at 46 pounds pressure is 9.018 cu. ft. 9.403 − 9.018 = .385 cu. ft., the difference in volume for a difference in pressure of 2 lb. $\frac{.385}{2}$ = .1925 cu. ft., the difference in volume for a difference in pressure of 1 pound. .1925 × .7 = .135 cu. ft., the difference in volume for a difference in pressure of .7 pound. Therefore, 9.403 − .135 = 9.268 cu. ft. is the volume of one pound of steam at 44.7 pounds pressure.

9.268 × 14 = 129.752 cu. ft. Ans.

EXAMPLE 5.—Find the weight of 40 cubic feet of steam at a temperature of 254° F.

SOLUTION.—The weight of one cubic foot of steam at 254.002°, from the table, is .078839 pound. Neglecting the .002°, the weight of 40 cu. ft. is, therefore, .078839 × 40 = 3.15356 pounds. Ans.

EXAMPLE 6.—How many pounds of steam at 64 pounds pressure absolute are required to raise the temperature of 300 pounds of water from 40° to 130° F., the water and steam being mixed together?

SOLUTION.—The number of heat units required to raise one pound from 40° to 130° is 130° − 40° = 90 B. T. U. (Actually a little more than 90 would be required, but the above is near enough for all practical purposes.) Then, to raise 300 pounds from 40° to 130° requires 90 × 300 = 27,000 B. T. U. This quantity of heat must necessarily come from the steam. Now, one pound of steam at 64 pounds pressure gives up, in condensing, its latent heat of vaporization, or 905.9 B. T. U.; but, in addition to its latent heat, each pound of steam on condensing must give up an additional amount of heat in falling to 130°. Since the original temperature of the steam was 296.805° F. (see table), each pound gives up by its fall of temperature 296.805 − 130 = 166.805 B. T. U. Consequently, each pound of the steam gives up a total of

905.9 + 166.805 = 1,072.705 B. T. U.

It will, therefore, take $\frac{27,000}{1,072.705} = 25.17$ pounds of steam to accomplish the desired result.

EXAMPLES FOR PRACTICE.

1. How many foot-pounds of work are required to change 42 pounds of water at 319° into steam at a temperature of 330°?
Ans. 29,190,679.8 ft.-lb.

2. How many B. T. U. are required to convert 25 pounds of water at 32° into 109.6 cubic feet of steam? Ans. 29,541.15 B. T. U.

3. Find the number of heat units required to change 11 pounds of water at 50° into steam at 100 pounds absolute pressure.
Ans. 12,802.526 B. T. U.

4. Find the weight of 712 cubic feet of steam at a pressure of 33 pounds gauge. Ans. 81.712 lb.

5. How many cubic feet of steam at 47.3 pounds pressure, gauge, are required to raise 120 pounds of water from 55° to 160° at atmospheric pressure? Ans. 82.359 cu. ft.

6. Find the volume of 19 pounds of steam at a temperature of 274°.
Ans. 175.883 cu. ft.

1207. The crude apparatus used in the first experiment above described (the cylinder containing the water and the piston, surrounded by fire) constitutes the simplest form of a steam engine. By means of it, heat may be transformed into work. It is certainly a very poor and inconvenient form of engine, since the only work it can do is that of raising weights; and, again, the fire must be successively removed and renewed every time a weight is lifted.

It will be seen that this engine consists of four essential elements; viz.: the fire, or source of heat; the water, or medium for the transfer of heat; the cylinder, with a piston to prevent the steam from escaping into the air, and a source of cold. Nearly all engines in actual operation consist of the above four elements, more advantageously arranged. In the actual engine, the fire is not in contact with the cylinder, but the steam is generated in a separate vessel known as the **boiler.** Again, the cylinder is of a fixed length, and the steam usually acts on both sides of the piston, instead of on one side.

This improvement of the actual engine over the simple apparatus of the first experiment leads to the consideration of some further essential characteristics of steam.

EXPANSION OF STEAM.

1208. Consider a closed cylinder (Fig. 236) containing a piston. The left end of the cylinder is open to the entrance of steam from the boiler, and the right end is open to the atmosphere, so that the steam on the right side of the piston may freely escape. The pressure of the steam from the boiler will drive the piston to the right end of the cylinder, pushing the steam on the right of the piston into the atmosphere. When the piston arrives at the right of the cylinder the right end is put in communication with the boiler and the left end with the atmosphere; the piston then moves back to the left end. The operation is thus repeated indefinitely.

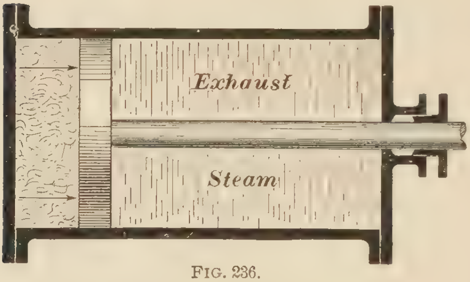


FIG. 236.

1209. Suppose that, during the entire movement of the piston from left to right, the left end of the cylinder is in communication with the boiler, and, further, that the steam on the right of the piston escapes freely into the atmosphere without requiring any work on the part of the piston to push it out. We wish to know the relation between the pressure and the volume at all points of the stroke, and the work done by the steam. The relation between the pressures and volumes may be shown graphically as follows: Draw two lines, $O\,X$ and $O\,Y$, at right angles to each other, as shown in Fig. 237. (Refer also to Fig. 223.) Then, the volume of the steam for any position of the piston may be represented by some distance on the line $O\,X$, and likewise the pressure of the steam at any position of the piston may be represented by the length of a vertical line parallel to $O\,Y$. Suppose that the area of the piston in Fig. 236 is 2 square feet, and that the distance moved through by the piston is 6 feet.

The length *O 3* of the diagram then represents a piston travel of 6 feet, or a steam volume of $6 \times 2 = 12$ cu. ft. Likewise, *O 1* represents a piston travel of 2 ft. and a steam volume of 4 cu. ft., and *O 2* a piston travel of 4 ft., and a steam volume of 8 cu. ft.

Now, through *1*, *2*, and *3* draw lines parallel to *O Y*, and on these lines lay off the length *1-1'*, *2-2'*, *3-3'*, representing, respectively, the pressures of the steam at the volumes *O 1*, *O 2*, and *O 3*. Connecting the points *1'*, *2'*, and *3'* by

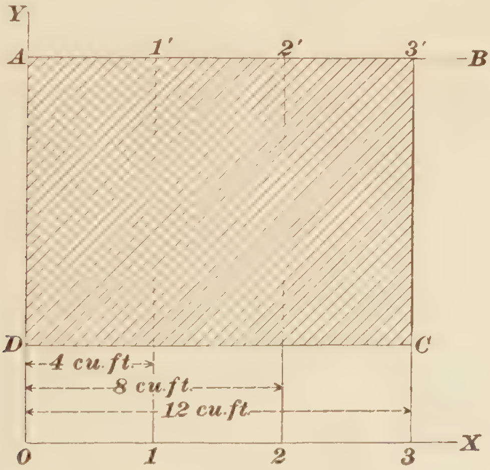


FIG. 237.

a line, we have a complete graphical representation of the relation between the pressures and volumes.

1210. Since the left end of the cylinder is in communication with the boiler during the whole motion of the piston from left to right, and since the temperature and pressure of the steam in the boiler is constant, it follows that the temperature and pressure of the steam in the cylinder will be the same for all positions of the piston. Suppose the pressure of the steam, when the piston is just starting from the left end of a cylinder, to be 60 pounds per square inch absolute. Let 1 inch on the

diagram represent a pressure of 30 pounds. Then, the length $O\ A$, which represents the steam pressure in question, must have a length of $\frac{60}{30} = 2$ inches; and, since the pressure is constant for all positions of the piston, the lengths *1-1′*, *2-2′*, *3-3′*, *4-4′*, etc., must all be equal to 2 inches. In other words, the line of pressures $A\ B$ must be parallel to the line $O\ X$. It was observed above that the temperature of the steam is also constant throughout the stroke of the piston. Hence, the above line $A\ B$ is the *isothermal* (constant temperature line) of saturated steam. Hence, it follows that *the isothermal of saturated steam is a straight line parallel to the axis of volumes $O\ X$.*

1211. The pressure $O\ A$ on the left of the piston is taken from vacuum. There is also a pressure on the right of the piston of 14.7 pounds per square inch, since the right end of the cylinder is open to the atmosphere. Let this be represented by the height $O\ D = \frac{14.7}{30} = .49$ in. Since this atmospheric pressure is constant throughout the stroke of the piston, it may be represented by the straight line $D\ C$ parallel to $O\ X$. The *net pressure* on the piston is then represented by

$$O\ A - O\ D = D\ A = 60 - 14.7 = 45.3 \text{ lb. per sq. in.}$$

The work done by the steam during one stroke of the piston may now be easily found.

There is a constant net pressure throughout the stroke of 45.3 lb. per sq. in. $= 45.3 \times 144 = 6{,}523.2$ lb. per sq. ft. The total pressure on the piston is, therefore, $6{,}523.2 \times 2 = 13{,}046.4$ lb. Work is defined as pressure, force, or resistance, multiplied by the distance through which it acts. Consequently, the work spent in moving the piston from one end of the cylinder to the other, a distance of 6 feet, is $13{,}046.4 \times 6 = 78{,}278.4$ foot-pounds.

1212. For our purpose, the work may be obtained in a more convenient way. As shown above, the pressure on the piston per square foot is 6,523.2 pounds. The volume of the

cylinder = area × length = 2 sq. ft. × 6 ft. = 12 cu. ft. Now, multiplying the pressure in pounds per square foot by the volume of the cylinder in cubic feet, the result is 6,523.2 × 12 = 78,278.4 foot-pounds, as before.

Let P = the pressure on the piston in lb. per sq. ft.;
p = pressure on piston in lb. per sq. in.;
V = volume in cubic feet swept through by the piston;
W = work in foot-pounds.

Then, $$W = P\,V = 144\,p\,V. \qquad \textbf{(93.)}$$

Referring to Fig. 237, it will be remembered that the line $D\,A$ represents the pressure p, and the line $D\,C$ represents the volume V. The product $D\,A \times D\,C$ = area of the rectangle $A\ B\ C\ D$. But $A\ D \times D\ C = p\,V$. *Therefore, the area of the rectangle is proportional to the work of the steam.*

$D\,A = \frac{45.3}{30} = 1.51$ inches. $D\,C = 2$ inches. Then, the area $A\ B\ C\ D = 1.51 \times 2 = 3.02$ sq. in. But 1 inch in height = 30 lb. per sq. in., and one inch in length $\frac{12}{2} = 6$ cu. ft. Hence, $p\ V = 3.02 \times 30 \times 6 = 543.6$ foot-pounds, and $W = 144\,p\,V = 543.6 \times 144 = 78{,}278.4$ foot-pounds.

1213. In the same way it can be shown that, no matter what the form of the area $A\ B\ C\ D$ may be, it will represent the work done by the steam, provided the upper line $A\ B$ represents the relation between the pressures and volumes of the steam on one side of the piston, and the lower line $C\ D$ represents the relation between the pressures and volumes on the other side. For, as shown in Art. **1158**, the area is equal to the length of the diagram multiplied by the mean ordinate, and the mean ordinate simply represents the average net pressure on one side of the piston.

Hence, in general, to find the work of the steam from the diagram, *multiply the area in square inches by the vertical scale of pressures, by the horizontal scale of volumes, and by 144. The result is the work in foot-pounds.*

EXAMPLE.—The area of a diagram of the character shown in Fig. 237 is 7.34 sq. in. The vertical scale of pressures is 36 pounds to the inch, and the horizontal scale of volumes is $2\frac{1}{2}$ cu. ft. to the inch. What is the work of the steam per stroke of piston?

SOLUTION.—Work $= 7.34 \times 36 \times 2\frac{1}{2} \times 144 = 95{,}126.4$ ft.-lb.

1214. Suppose now that the left end of the cylinder is shut off from the boiler just as the piston in Fig. 237 has completed $\frac{1}{3}$ of the stroke—that is, when it is just 2 feet from the left end of the cylinder. The pressure on the left of the piston, as in Fig. 237, will be constant so long as the cylinder is in communication with the boiler—that is during

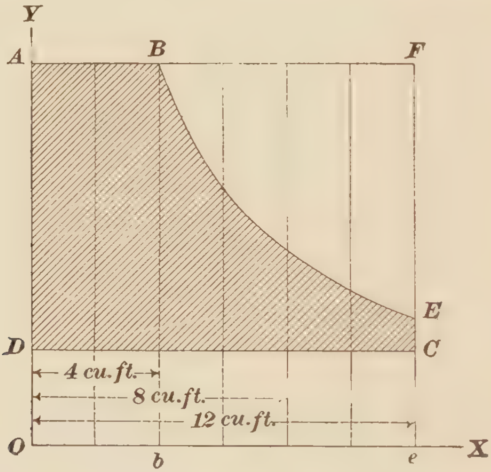


FIG. 238.

the first 2 feet of the stroke. Therefore, the pressure line for the first $\frac{1}{3}$ of the diagram will be a straight line $A\ B$ parallel to $O\ X$ (see Fig. 238), and, as before, $\frac{60}{30} = 2$ inches above it. At this point the steam supply is shut off, and the 4 cubic feet of steam in the cylinder must expand as the piston moves along, until it fills the entire volume of the cylinder. Its pressure, then, must fall somewhat after the manner of a perfect gas. This fall in pressure is graphically represented by the curved line $B\ E$. Assuming for the moment that the curve is the *equilateral hyperbola* (see Art.

1161), the work may be calculated as follows: The height of Fig. 238 is 2 inches; the length $A\,B$ is $\frac{2}{3}$ inch. Therefore, the area $A\,B\,b\,O = 2 \times \frac{2}{3} = 1.3333$ sq. in.

1215. The area $B\,E\,e\,b$ may be found by the following formula, in which

A = area of diagram in square inches;
p_1 = initial pressure (as $O\,A$) measured in inches;
V_1 = volume at cut-off, measured in inches.

$$A = 2.3026 \times p_1\,V_1 \log \frac{V_2}{V_1} \qquad \textbf{(94.)}$$

$$= 2.3026 \times \tfrac{2}{3} \times 2 \times \log \frac{6}{2} = 1.4648 \text{ sq. in.}$$

Then, area $A\,B\,E\,e\,O = 1.3333 + 1.4648 = 2.7981$ sq. in.
$O\,D = .49$ in. Area $C\,D\,O\,e = .49 \times 2 = .98$ sq. in.
Area $A\;B\,E\,C\,D = A\;B\,E\,e\,O - C\,D\,O\,e = 2.7981 - .98 = 1.8181$ sq. in.

Then, $L = 144\,a\,h\,h_1$, **(95.)**

in which, L = work in foot-pounds;
a = net area of diagram (as $A\,B\,E\,C\,D$);
h = scale used to lay off pressures;
h_1 = scale used to lay off volumes, the volumes being always expressed in cubic feet, and the pressures in pounds per square inch.

In the above diagram

$L = 1.8181 \times 30 \times 6 \times 144 = 47{,}125.15$ ft.-lb.

The work done in the first case was 78,278.4 ft.-lb.; so by shutting off the steam at $\frac{1}{3}$ of the stroke the work of the steam has been decreased by $78{,}278.4 - 47{,}125.15 = 31{,}153.25$ ft.-lb. But, in the first case, 12 cu. ft. of steam at boiler pressure were used, while, in the last case, only 4 cu. ft. were used; hence, the work per cubic foot of steam was, in the first case, $\frac{78{,}278.4}{12} = 6{,}523.2$ ft.-lb., and, in the second case, $\frac{47{,}125.15}{4} = 11{,}781.29$ ft.-lb., or nearly twice as much. This shows the economy of cutting off the steam early in the stroke and allowing it to expand.

EXAMPLES FOR PRACTICE.

1. The mean ordinate of a diagram, similar to that shown in Fig. 238, is 1.2 inches long. The vertical scale of pressures is 1″ = 40 lb. per sq. in., and the horizontal scale of distances is 1″ = 10″. The length of the diagram is 3″, and 1 ft. of actual length of the vessel which contains the steam represents a volume of 452 cu. in. What is the work done in one stroke of the piston? Ans. 4,520 ft.-lb.

2. The mean pressure in the cylinder of a steam engine is 38.7 lb. per sq. in.; the diameter of the cylinder, 26″; the stroke, 32″, and number of strokes per minute, 120. Find the horsepower by means of formula **93.** Ans. 199.244 H. P.

3. The mean ordinate of a diagram is .89″; the length, 3.2″; the vertical scale of pressure, 1″ = 50 lb. per sq. in.; the horizontal scale of volumes, 1″ (diagram) = .56 cu. ft. Find the work done in 12 strokes. Ans. 137,797.6 ft.-lb.

1216. The Expansion Line.—The character of the expansion curve $B\ E$, Fig. 238, depends upon a variety of circumstances. Suppose, first, that the steam expands without any condensation. The equation representing the relation between the pressures and volumes will then be

$$p\,V^{\frac{17}{16}} = \text{some constant.}$$

When the quantity of the expanding steam is one pound, the constant is 475, and the equation becomes

$$p\,V^{\frac{17}{16}} = 475,$$

which has already been given. The curve may be plotted by laying off the volumes, taken from column 7 of the steam table, along the line $O\ X$, and the corresponding pressures from column 1, along the line $O\ Y$. It will now be necessary to determine the conditions under which the steam will follow the above expansion line.

1217. Suppose that one pound of steam at 60 pounds pressure (absolute) be confined in a non-conducting cylinder. Its volume, according to the table, is 7.024 cubic feet. Let the pound of steam expand until the pressure reaches 54 pounds absolute; the volume, then, is 7.756. It may be assumed with little error that the portion $a\ b$, Fig. 239, of the expansion line, which represents the drop of pressure from 60 to 54 pounds, is straight. The average pressure is, then, $\dfrac{60 + 54}{2} = 57$ pounds.

The volume $d\,c$ is $7.756 - 7.024 = .732$ cu. ft.

The work $= 144\,p\,V = 144 \times 57 \times .732 = 6{,}008.256$ ft.-lb.

The total heat of the steam at 60 pounds pressure is 1,171.176 B. T. U. per pound, and at 54 pounds pressure, 1,169.102 B. T. U. per pound. Hence, the pound of steam during its expansion gives up $1{,}171.176 - 1{,}169.102 = 2.074$ B. T. U. $= 1{,}613.572$ foot-pounds.

1218. The work done by the expanding steam is 6,008.256 foot-pounds, while the work given up by the steam during its expansion is only 1,613.572 foot-pounds, or a little over $\frac{1}{4}$ as much. It is evident that the work could not have

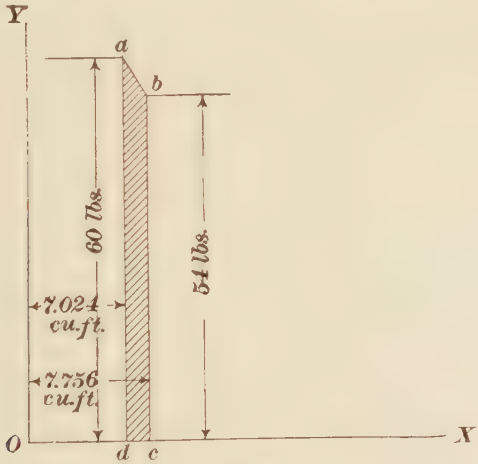


FIG. 239.

been done by the expansion of the steam alone. There remains $6{,}008.256 - 1{,}613.572 = 4{,}394.684$ foot-pounds of work that must have been obtained in some other manner. The cylinder is non-conducting, so no heat could have been obtained from without. The *only* way in which this extra amount of work can be accounted for is to suppose that a small amount of steam is condensed while expanding, and thus gave up its latent heat of vaporization. 4,394.684 foot-pounds $= 5.649$ B. T. U. The average pressure was 57 pounds, and the latent heat of steam at that pressure is

911.304 B. T. U. Hence, $\frac{5.649}{911.304} = .0062$ of the pound of steam must have condensed to furnish the extra 4,394.684 foot-pounds of work. We thus arrive at the conclusion that *saturated steam expanding in a non-conducting cylinder must of necessity condense somewhat.*

In the same manner it can be shown that, if saturated steam be compressed in a non-conducting cylinder, it must become superheated.

1219. It is thus apparent that steam in a non-conducting cylinder cannot expand according to Rankine's formula $p V^{\frac{17}{16}} =$ a constant. The curve is really the adiabatic for steam, and is usually represented by the formula,

$$p\, V^{\frac{10}{9}} = \text{a constant.}$$

The cylinder of the actual engine is never perfectly non-conducting. In fact, it is made of cast iron, which is a very good conductor of heat. This fact leads to great complication in the study of the action of the steam in the cylinder.

1220. Suppose, first, that the cylinder be simply exposed to the atmosphere—that is, it is not covered with any non-conducting material. The steam, entering one end of the cylinder with a pressure of say 80 pounds absolute, has a temperature of about 312°. As it meets the cold cylinder walls, it will raise them to 312°, nearly; but, at the same time, the steam, meeting a surface cooler than itself, must partially condense. At some point near the middle of the stroke, the communication with the boiler will be cut off, and the steam will expand, the pressure lowering to say 20 pounds absolute, and the temperature, consequently, lowering to about 228°. Therefore, at the end of the stroke, the cylinder walls will be at a temperature of about 228°. The fresh steam now enters the other end of the cylinder with a pressure of 80 pounds and a temperature of 312°, comes in contact with the cylinder walls, and partially condenses, this action being repeated at every stroke. In addition to this condensation, due to the alternate heating and cooling of the cylinder walls, there is the condensation which must always

accompany expansion, even in a non-conducting cylinder. Of course, this condensation is almost a total loss, since the water can exert no pressure on the piston.

Some of the condensed steam, however, may be, and generally will be, evaporated again towards the end of the stroke. Just before the steam supply is cut off, the temperature of the cylinder walls is 312°; as the steam expands, it partially condenses and its temperature falls. The temperature of the cylinder walls also falls, and in doing so, gives up heat, part of which is radiated and lost, and part re-evaporates some of the condensed steam. As a consequence of this action, the real expansion line falls below the theoretical line ($p\ V^{\frac{17}{16}} = constant$) just after cut-off, but rises above it towards the end of the stroke.

1221. A very common way of avoiding the loss due to the condensation of steam in the cylinder is to surround the latter with a **steam jacket**—that is, to keep the cylinder continually immersed in steam at boiler pressure. Heat is then transferred from the steam outside the cylinder to the expanding steam inside, and condensation is thus prevented, or, rather, it takes place in the jacket instead of in the cylinder. The expansion curve in this case approaches more nearly the theoretical curve, but usually rises a little above it towards the end of the stroke.

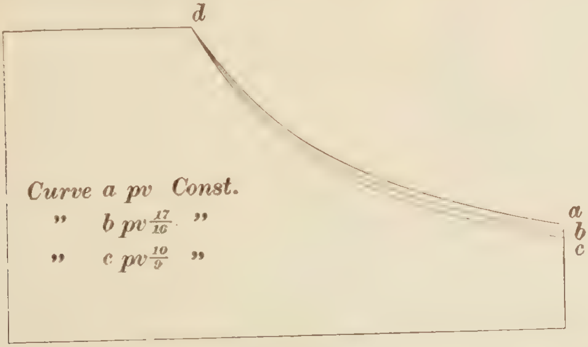


FIG. 240.

1222. Fig. 240 shows the three curves above considered. The upper one is the **equilateral hyperbola**, $p\ V = a$

constant; the second is the **curve of constant steam weight,** $p\,V^{\frac{17}{16}} = a$ *constant*, and the third is the **adiabatic** for steam, $p\,V^{\frac{10}{9}} = a$ *constant.*

It will be seen how marked may be the difference between the expansion curves under varying circumstances. It is usual, in practice, to assume that the expansion curve is the equilateral hyperbola. And, though it is in no sense the theoretical curve, it is generally nearer than the others, and rather easier to deal with. Hence, in all farther discussions, it will be assumed that expanding steam follows the equilateral hyperbola.

THE STEAM-ENGINE MECHANISM.

THE FOUR-LINK SLIDER CRANK.

1223. It has been shown how steam may do work by lifting weights against the pressure of the atmosphere; since, however, it is not generally desired to do work in this manner, it is essential that some method of changing the to-and-fro motion of the piston into a continuous motion in one direction should be devised. The form of mechanism used for this purpose in practically all types of engines is shown in

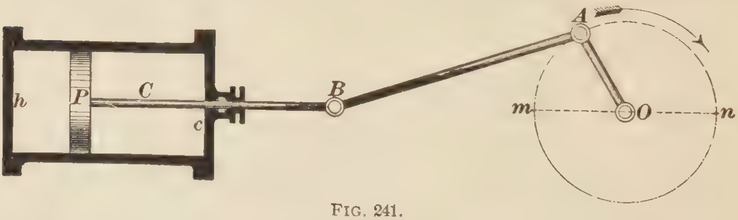


FIG. 241.

Fig. 241. It is technically known as the **four-link slider crank.** The steam from the boiler enters one end—say, in this case, the end h—of the cylinder, and pushes the piston to the other end. By means of another mechanism, called the **valve,** the steam is now admitted to the end c of the cylinder, while the end h is at the same time allowed to communicate with the atmosphere or with a condenser.

The steam in h escapes, while that in c pushes the piston back again to its original position, whence the same operation is repeated.

Attached to the piston, and forming a part of it, is the piston rod $C\,B$; to the end of $C\,B$ is fastened, by a joint, one end of the link $B\,A$. The other end of $B\,A$ is joined to the link $A\,O$; and the other end of $A\,O$ terminates in a shaft O, which is fixed in stationary bearings. It is evident that the end of $B\,A$, which is attached to $C\,B$, can move only in a straight line; and since the shaft O can rotate only in its bearings, the end of $A\,O$, which is attached to $B\,A$, can move only in a circle.

When the piston is at one extreme end of the cylinder, say at h, the joint A is at the point m, and all three links, $A\,O$, $B\,A$, and $C\,B$, lie in a straight line. As the piston moves to the right, the link $B\,C$ moves also to the right, while the joint A is constrained to move in the upper semicircle $m\,n$; when P arrives at the other end of the cylinder, the joint A is at n, and again $A\,O$, $B\,A$, and $C\,B$, are in a straight line. The piston now moves back to the end h of the cylinder, the joint A moving in the lower semicircle from n to m.

1224. The link $A\,O$ is called the **crank**, $B\,A$ the **connecting-rod**, and $C\,B$ the **piston rod.** Those parts which have a to-and-fro or reciprocating motion are called the **reciprocating parts.**

The end h of the cylinder is called the **head end,** and c the **crank end.** The distance passed through by the piston during a half revolution of the crank is called the **stroke;** the stroke is evidently equal to the diameter $m\,n$ of the circle described by the end a of the crank.

The engine may run in the direction shown by the arrows in the figure, or it may run in the reverse direction. In the former case it is said to **run over,** and in the latter case to **run under.**

The stroke from the head to the crank end of the cylinder—that is, from left to right in the figure—is called the **for-**

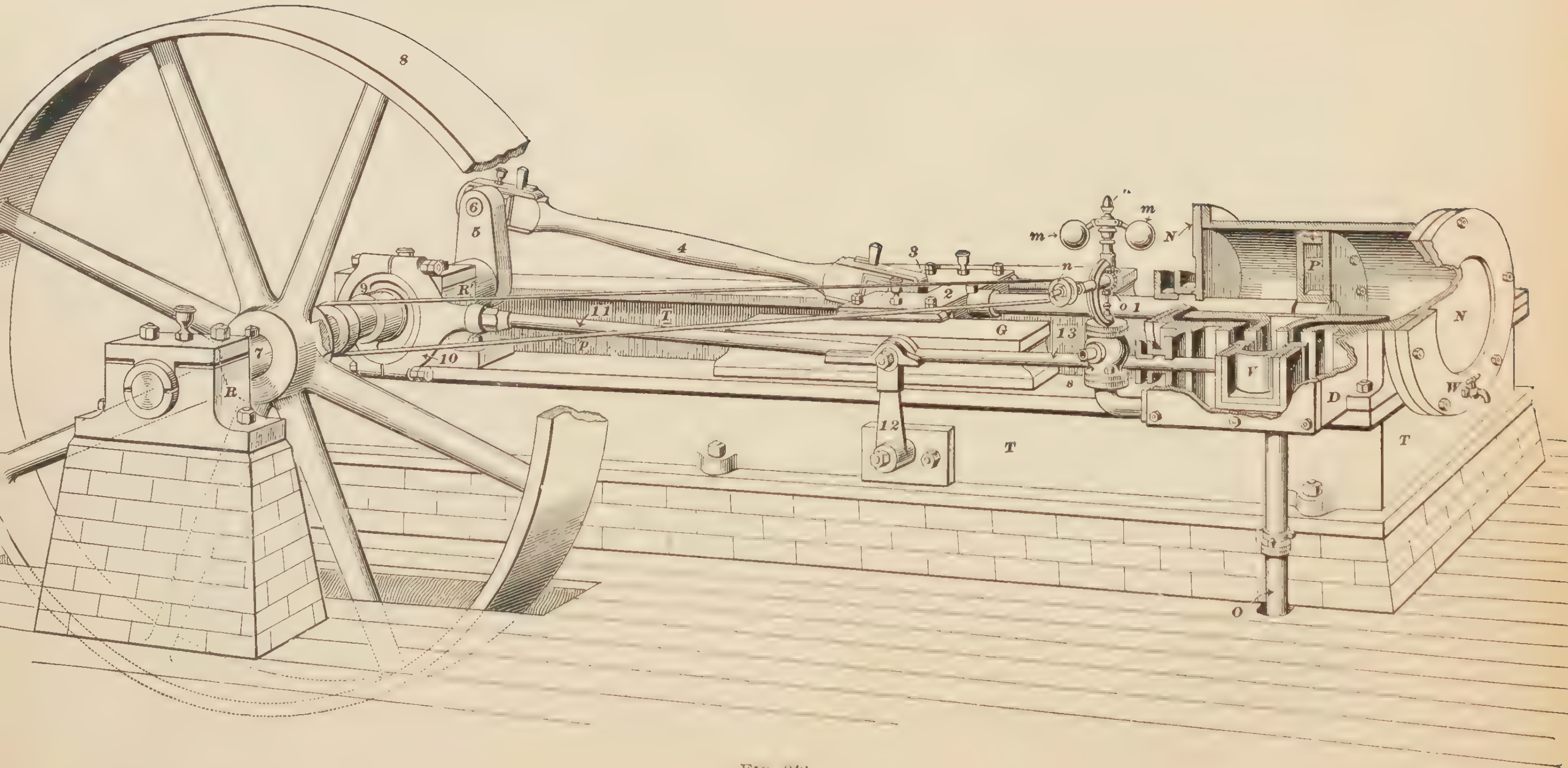


FIG. 242.

ward stroke; the one from crank end to head end, the **return stroke.**

The above simple mechanism perfectly fulfils the office of giving a continuous motion in one direction. A pulley is keyed to the shaft O, and the power is transferred by belting from the pulley to shafting, or directly to the machinery to be run.

THE PLAIN SLIDE-VALVE ENGINE.

1225. The enormous number of types of the steam engine may be classified

(1) According to the kind of service, as	Stationary, Locomotive, Marine, etc.
(2) According to number and arrangement of cylinders, as	Simple, Compound, Triple expansion, Quadruple expansion, Duplex, etc.
(3) According to the type of valve used, as	Plain slide-valve, Automatic cut-off, Corliss, etc.

They may all be *horizontal* or *vertical*, *condensing* or *non-condensing*, *single-acting* or *double-acting*. All of these different types involve essentially the same principles, and, therefore, the description of a single type will be sufficient to give a general knowledge of these principles. For this purpose we shall choose the **simple slide-valve engine.**

1226. In Fig. 242 such an engine is shown, and in Fig. 243 is shown a section of a steam cylinder. Referring to these figures, H is the head end, and C the crank end of the steam cylinder; B and B' are the steam ports; D is the steam chest; E is the exhaust port; N and N' are the cylinder heads; S is the steam supply pipe; O is the exhaust pipe, and connects with the exhaust port E; G is one of the two guide bars; R and R' are the shaft bearings, and T is the bed, or frame, of the engine. The above are

all stationary parts of the engine, or parts which do not change their relative positions when the engine is in motion. *P* is the piston; *1* is the piston rod; *2* is the cross-head; *3* is the cross-head pin; *4* is the connecting-rod; *5* is the crank; *6* is the crank-pin; *7* is the crank-shaft; *8* is the fly-wheel; *9* is the eccentric; *10* is the eccentric-strap; *11* is the eccentric rod; *12* is the rocker; *13* is the valve rod or stem, and *V* is the slide-valve. These are all movable parts of the engine, or parts which change their relative positions when the engine is in motion. *W* is the working length of

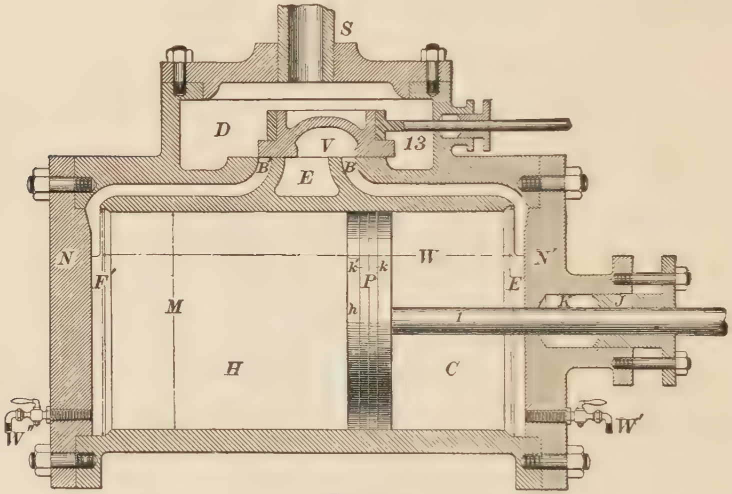


FIG. 243.

the cylinder. It is slightly less than the distance between the cylinder heads, since a small space must be left between the head and the piston, when the latter is at the end of its stroke. This space, together with the volume of the steam port, which leads to it, is called the **clearance.**

1227. The **stroke** of the engine is the travel of the piston *P*; since the piston and cross-head are rigidly fastened to the same rod, the stroke must also be equal to the travel of the cross-head. It was shown in Fig. 241 that the stroke is also equal to the diameter of the circle described by the crank-pin *6*, or, what is the same thing, it is equal to

twice the length of the crank *5*, this length being measured from the center of crank-pin *6* to the center of the crank-shaft *7*. *M* is the diameter or **bore** of the cylinder.

1228. The size of an engine is generally expressed by giving the diameter of the cylinder and the stroke in inches. Thus, an engine having a cylinder diameter of 16 inches, and a stroke of 22 inches, is called a $16'' \times 22''$ engine.

1229. At the ends *F* and *F'* the cylinder is **counter-bored**—that is, for a short distance, the bore is greater than *M*. The piston projects partly into this counterbore at the end of each stroke. Were it not for the counterbore, the piston would not wear the cylinder walls their entire length, and shoulders would be formed at each end of the cylinder. When the wear of the joints in the connecting-rod is taken up, the length of the connecting-rod is slightly increased, and the piston is shoved back slightly towards the head end of the cylinder. In this case, the shoulder would cause an undesirable pounding of the piston.

Drain-cocks *W'* and *W''* are fitted in each end of the cylinder, through which any condensed steam may be discharged.

1230. The piston fits loosely in the cylinder and has split rings *k* and *k'* inserted, which spring out so as to press against the wall of the cylinder, and prevent leakage of steam between the wall of the cylinder and piston. Pistons are usually supplied with a follower-plate *h*, which is bolted to the head end of the piston *P*, in order to hold these split rings *k* and *k'* in place. The piston rod *1* is a round bar, rigidly connected to both the piston *P* and the cross-head *2*.

K is a stuffing-box in which packing is placed, and is fitted with a gland *J*, which, when bolted down, compresses the packing around the piston rod *1*, and makes a steam-tight joint. This packing is usually made in the form of split rings, which are so placed that the split of the first ring is covered by the solid part of the next ring. When repacking, care should be taken not to cause unnecessary friction by

too much pressure from the gland. The cross-head *2* is given an easy sliding fit between the guide bars *G* and *G'*, Fig. 242, which are in line with the path of the piston rod, and combine with the cross-head to relieve the piston rod of all bending strains.

The connecting-rod *4* forms the connecting link between the cross-head and crank *5*. The joint between cross-head *2* and connecting-rod *4* is made by the cross-head pin *3*; and the joint between the connecting-rod and crank is made by the crank-pin *6*. Connecting-rods are usually made from 4 to 6 times the length of the crank, or from *4 to 6 cranks* in length, as it is called.

1231. The eccentric *9*, which imparts motion to the slide-valve *V*, is the exact equivalent of a crank having the

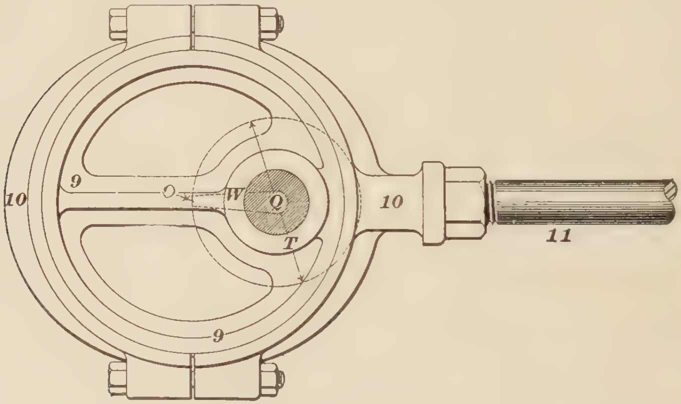


FIG. 244.

same throw. This is clearly shown in Fig. 244, in which *9* is the eccentric, *10* is its strap, *11* is the eccentric rod, and *W* is a crank having a throw *T* equal to that of the eccentric. The center of the crank-shaft *Q* is the center of rotary motion. The dotted circle represents the path of the common center *O* of the eccentric and crank *W*. The eccentric revolves freely in the eccentric strap *10*, which is rigidly connected to the eccentric rod *11* (see Figs. 242 and 244). In practice, the diameter of the shaft generally exceeds the throw of the eccentric. In plain slide-valve

engines, the eccentric is usually keyed to the shaft after being properly adjusted. The connection between the eccentric rod *11*, Fig. 242, and the valve stem *13*, is accomplished in a variety of ways. In the figures, a rocker arm *12* is used to support the joint between the eccentric rod *11* and the valve stem *13*. The latter must be supported in some such manner to prevent it from binding in its stuffing-box.

THE D SLIDE-VALVE AND STEAM DISTRIBUTION.

1232. Of the different kinds of valves used to distribute the steam in the engine cylinder, the **D** slide-valve is the

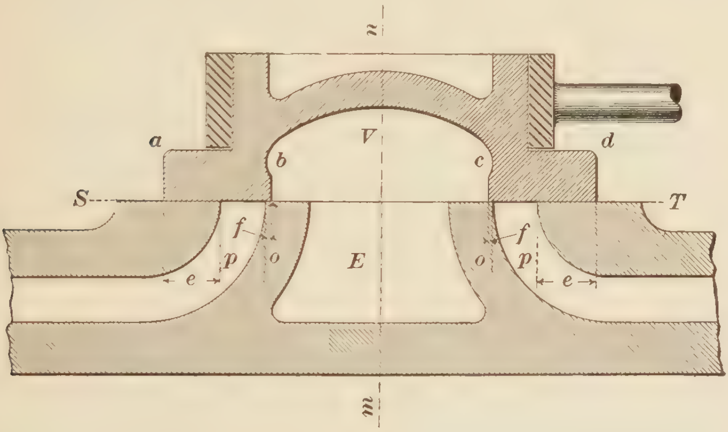


FIG. 245.

most common. A section of such a valve is shown in Fig. 245, in its central position; *p* and *p* are the **steam ports,** *o* and *o* the **bridges,** *E* the **exhaust port,** *S T* the **valve seat.** The flanges of the valve *a b* and *c d* are seen to be wider than the ports which they cover. Of this extra width, the parts *e*, *e* are called the **outside lap,** and the parts *f*, *f* the **inside lap.** The valve is here shown in mid position; i. e., the center line *n* of the valve coincides with the center line *m* of the exhaust port. As the motion of the valve is given by the eccentric, the valve is in mid position when

the radius of the eccentric ($Q\ O$, Fig. 244) is in a vertical position. When $Q\ O$ lies horizontally on the right side of Q, the valve (see Fig. 242) is in its position nearest the head end of the steam chest, and when $Q\ O$ lies horizontally on the left side of Q, the valve is at the end of its stroke in the crank end of the steam chest. In order to show how the steam is distributed in the cylinder by means of the valve and eccentric, a series of skeleton diagrams have been drawn showing the relative positions of the valve and piston for different points of a double stroke.

1233. Fig. 246 shows five diagrams. They represent a **D** slide-valve without lap or lead. $O\ a$ represents the crank; $O\ b$ the eccentric (which was shown to be equivalent to a crank); $a\ c$ the connecting-rod, and $b\ d$ the eccentric rod. It should be remarked that the sizes of the various parts have been greatly exaggerated, particularly the radius of the eccentric circle, and the amount of clearance. Diagram A, Fig. 246, represents the piston as just on the point of beginning the forward stroke. The valve is moving in the direction of the arrow, and the outer edge is just about to admit steam to the left-hand port. As will be seen, the valve is in its central position, and, consequently, the line joining the center of the shaft and the center of the eccentric (this line will hereafter be called *eccentric radius*) is vertical. All of the parts are about to move in the direction of the arrows. Diagram B shows the position of the parts when the crank has moved through 90° from its position in A. The piston is at the middle of its stroke, or very nearly there. It would be exactly at the middle of its stroke, but for the fact that the connecting-rod makes an angle with the horizontal. The angularity of the connecting-rod will be treated of later; for the present, it will be assumed that it has no effect on the position of the piston. The valve has reached the extreme limit of its travel to the right, and the eccentric radius $O\ b$ is horizontal. The left steam port is fully opened for the live steam, and the right steam port is fully opened for exhaust. Another crank movement of 90°

places the different parts as shown in diagram C. The piston has reached the end of its forward stroke; the valve is in its central position moving towards the left, and has just

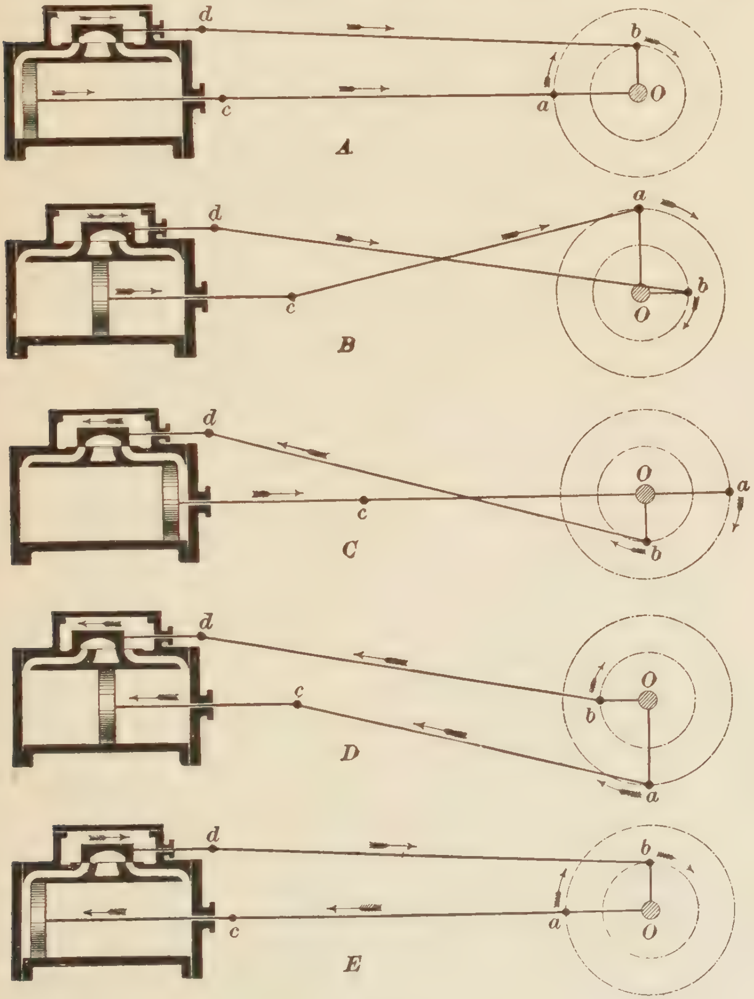


Fig. 246.

closed the left steam port and right exhaust port, and is just about to open the right port for admission of live steam,

and the left port for the release of exhaust steam. The piston has now traveled one full stroke. Diagram D shows the piston in its central position on the return stroke. The crank is in the position $O\ a$; the eccentric is horizontal, as represented by $O\ b$, and the valve is at the farthest point of its travel to the left, the right port being fully open for live steam, and the left port fully open for exhaust. In the diagram E, the piston has reached the extreme point of the return stroke, the piston rod, connecting rod, and crank being all in one straight line; this also occurs in diagrams A (which is the same as E) and C. The valve has been moving to the right, and is now in its central position, just on the point of admitting steam to the left port.

1234. These diagrams show conclusively that, with no lap or lead, the steam is admitted to the cylinder for the full stroke of the engine, consequently there can be no cut-off, and, therefore, no expansion of steam.

The following conclusion is now evident: *With an ordinary* **D** *slide-valve, operated by one eccentric, there can be no cut-off, and, therefore, no expansion of steam, unless the valve has outside lap.*

1235. The effect of lap on the movement of the valve relatively to the piston, and also on the movement of the eccentric and crank, is clearly shown in Figs. 247–254. In these figures, the valve has both outside and inside lap, but no lead. These diagrams have been distorted, as was done in Fig. 246, in order that the eccentric radius might be long enough to show up well. In Figs. 247–254 the eccentric radius is three times as long as it should be for the amount of valve movement shown by the figure. The diameter of the crank circle is also a little greater than the stroke of the piston for the same reason. In order to show the distribution of steam by the valve, a diagram has been drawn above and below each cylinder, those above being marked M, and those below, N. These diagrams are supposed to be drawn in the following manner: Imagine it to be possible to connect two small pipes to the piston, one on each side.

Suppose each pipe has a steam-tight piston working in it, the lower side of the pistons being subjected to the steam pressure in the cylinder, and the upper side to the atmospheric pressure. Suppose, further, that there is a coiled spring on top of the piston; that a piston rod passes through the center of the spring; and that a pencil is attached to the end of the piston rod. If a pressure of 10 pounds is required to compress the spring 1 inch, it is evident that for every 10 pounds pressure in the cylinder, the pencil will move upwards 1 inch, and, if it touched a sheet of paper, would mark a line on that paper. It will now be presumed that an arrangement like that just described is attached to the steam engine piston, and that the pencil touches a sheet of paper, which is held stationary. Then, when the steam piston moves ahead, the pencils will make straight lines at heights corresponding to the steam pressure on the under sides of the little pistons, except when the pressure of the steam in cylinder varies, in which case, the pencil will move up or down, according as the pressure increases or diminishes.

Having made these suppositions clear, let $Q X$, Figs. 247 to 254, represent the line which the pencil would trace if there were a perfect vacuum in the cylinder; i. e., $Q X$ is the line of no pressure; also, let $A B$ represent the line which the pencil would trace if the pressure in the cylinder was just equal to that of the atmosphere, and $Q Y$ the line of no volume. Then the point Q represents no volume and no pressure. Finally, let $I D$ represent the volume of the clearance.

1236. Consider Fig. 247. The piston is represented as just beginning the forward stroke, and the valve as just commencing to open the left steam port, both moving in the same direction, as shown by the arrows. If the valve had no outside lap, the position of the eccentric center would be at e, but on account of the lap, the valve has moved ahead of its central position in order to bring its edge to the edge of the port. To accomplish this, the eccentric center has

been moved from e to b, $O\ b$ being the position of the eccentric radius. The angle $b\ O\ e$, which the eccentric radius makes with the position it would be in if there were no lap or lead, is called the **angle of advance.**

Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance space on the left of the piston, so that the steam acts on the piston at full boiler pressure. If the length of the line $A\ 1$ represents the boiler pressure (gauge), the pencil which registers the pressure on the left side of the

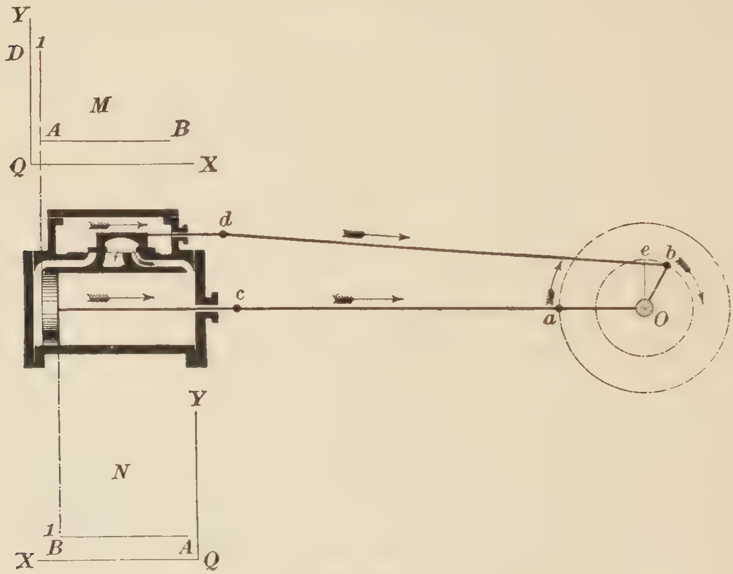


FIG. 247.

piston will be at *1*. The steam on the right side of the piston is flowing (*exhausting*) into the atmosphere through the exhaust port as shown by the arrow. As the size of the exhaust port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston in addition to the atmospheric pressure. This is termed **back pressure.** Therefore, in the diagram N, let *1* be the position of the second pencil, then, $1\ B$ is the back pressure.

1237. Fig. 248 shows the position of the piston and valve when the exhaust port is fully open. The crank has

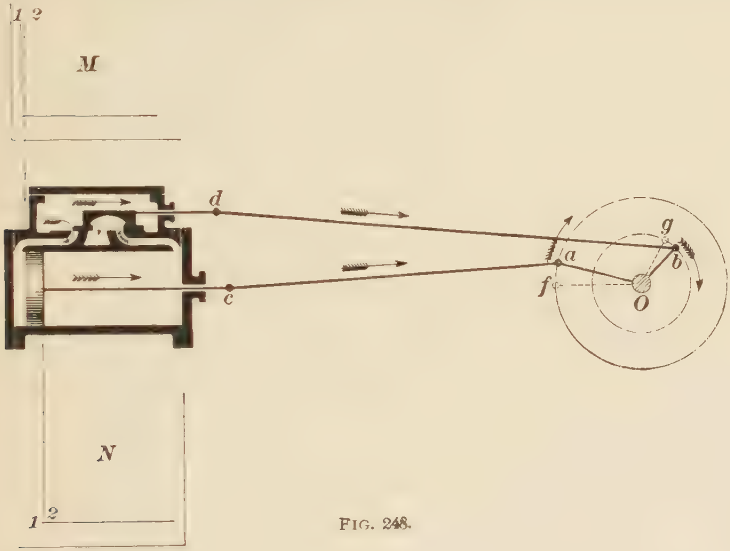


FIG. 248.

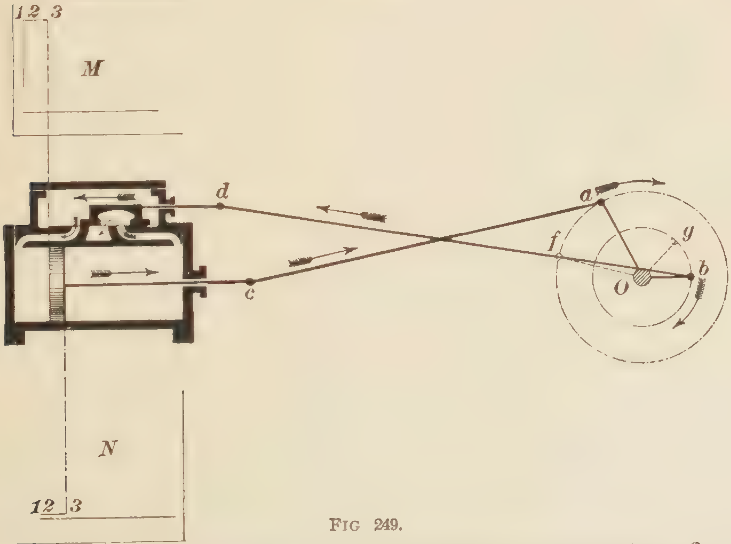


FIG 249.

moved from the position $O\ f$ (shown by dotted line) to $O\ a$, and the eccentric center from g to b. Steam is entering the head

end of the cylinder and exhausting at the crank end. The pencils have moved from *1* to *2* on both diagrams *M* and *N*.

1238. In Fig. 249 the piston has advanced far enough to enable the valve to reach the end of its stroke and open the port its full width. The crank and eccentric have moved to the positions *O a* and *O b*, the dotted lines showing their last position in Fig. 248. The eccentric radius is horizontal, and any further movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back. In the diagrams *M* and *N*, the pencil has traced the lines *1-3*.

1239. Fig. 250 shows the piston still further advanced on its stroke, and the valve as having its inside edge in line with the outside edge of the exhaust pipe. The left end of

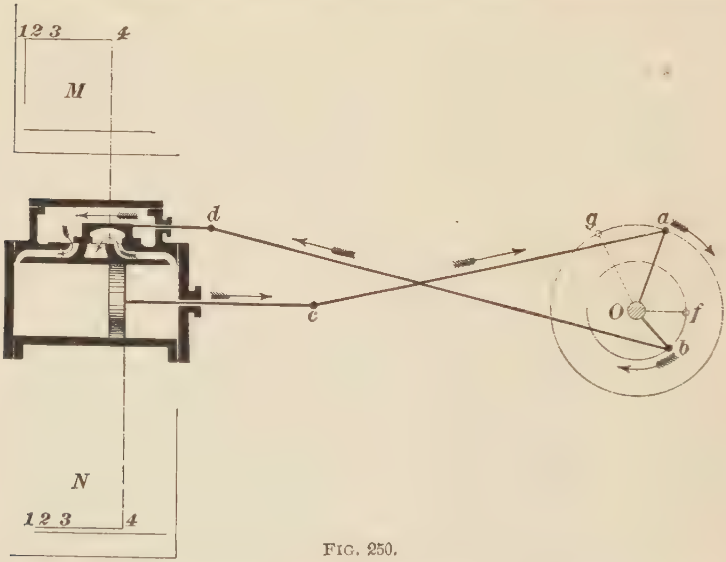


FIG. 250.

the valve has partially closed the steam port. The amount of advancement of the crank and eccentric from their last positions is shown by their distances from the dotted lines. The pencils have traced the lines *3-4* on the diagrams *M* and *N* during this movement of the piston from the last position.

Fig. 251 marks one of the most important points of the stroke. Here the valve has closed the steam port; i. e.,

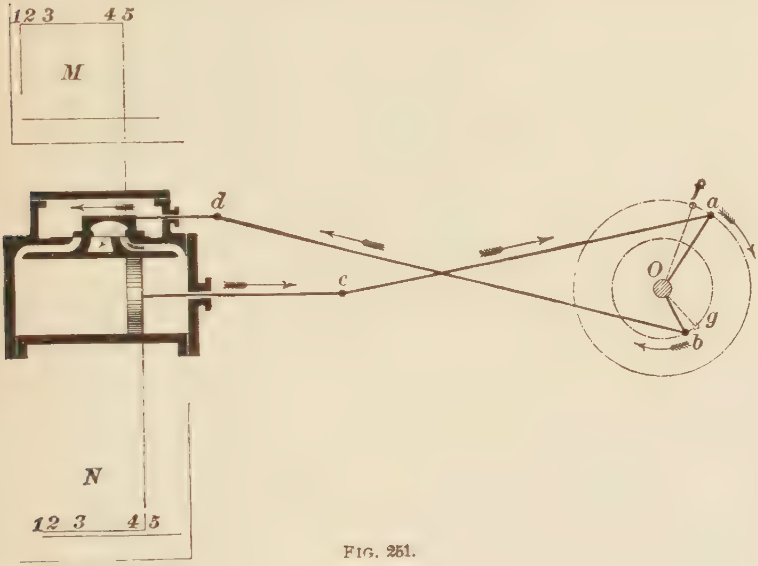


FIG. 251.

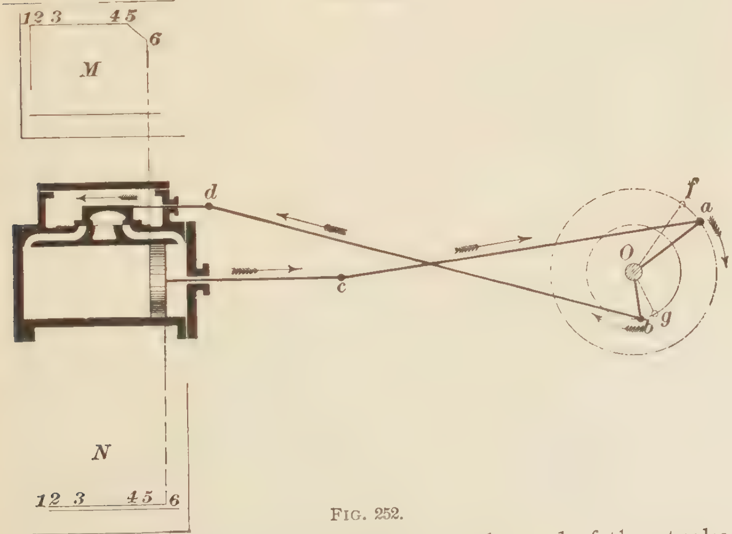


FIG. 252.

cut off the steam, and from here to the end of the stroke, the steam in the cylinder expands. The exhaust port is

now partially closed. The crank and eccentric have moved through the angles indicated. During this movement, the pencils have traced the lines *4-5*.

1240. Fig. 252 shows another very important valve position. Here the inside edge of the valve closes the exhaust port, and, from now on to the end of the stroke, the steam in front of the piston is compressed. In the diagrams *M* and *N*, the lines *5-6* are traced by the pencils. The line *5-6* on the diagram *M* is an expansion line, the pressure falling as the piston moves ahead.

1241. In Fig. 253, the piston has advanced far enough to cause the left inside edge of the valve to be in line with

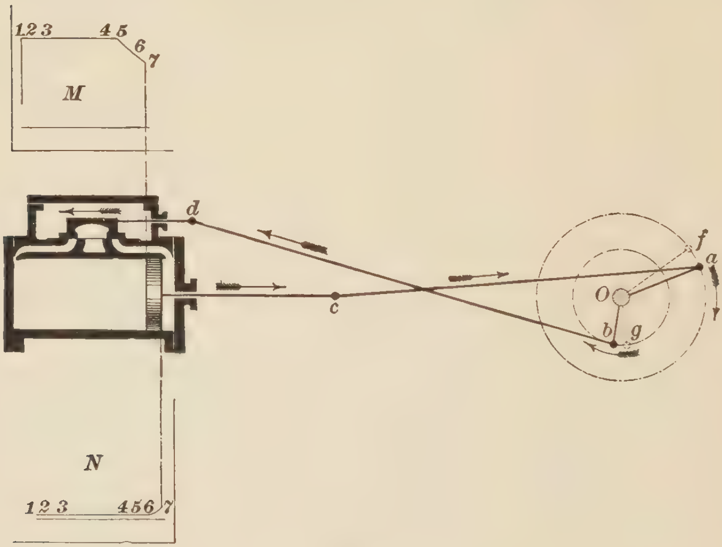


Fig. 253.

the inside edge of the left port. The slightest movement of the valve to the left will open the left port to exhaust. Expansion really ends here, although, on account of the limitation in the size of the ports, there will still be a slight further expansion owing to the inability of the steam to escape instantly. During this last movement of the

piston, the pencils trace the lines *6-7* on the diagrams *M* and *N*. On the diagram *M* the line *6-7* is a continuation of the expansion line *5-6*, while in the diagram *N* it shows part of the compression line, the pressure rapidly increasing as the piston nears the end of the stroke.

1242. In Fig. 254 the piston has reached the end of its forward stroke and is about to begin the return stroke. The right outside edge of the valve is in line with the outside edge of the right port. The steam is exhausting from the head end of the cylinder, as shown by the arrows. The

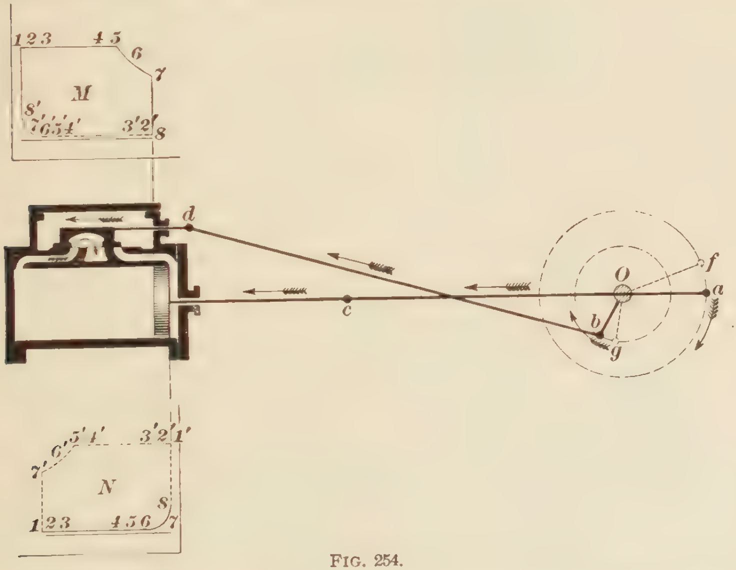


FIG. 254.

crank and eccentric are both diametrically opposite their positions in Fig. 247. In the diagrams *M* and *N*, the pencils have traced the lines *7-8*. *M* shows that the pressure has fallen very rapidly from *7* to *8*, while in *N* it has risen from *7* to *8*. The very slightest movement of the piston to the left will admit steam to the crank end of the cylinder and cause the pencil to rise to the point *1'*.

1243. During the return stroke the above described actions of the steam will be repeated, the pencils tracing the

dotted lines on the diagrams *M* and *N* in Fig. 254, the exhaust going through the left port and the steam through the right port. As the process is so nearly like the preceding, the diagrams have not been drawn, but the student should follow the valve through the different positions, and note the effects on the diagrams. To assist him in this, the corresponding points have been numbered as in the foregoing figures.

1244. Effects of Lap.—The study of figures 247 to 254 should show the effects caused by varying the lap. Thus, in Fig. 251, it is evident that if the outside lap had been less, the valve would not close the left port when its center was in the position shown; consequently, the piston must move further ahead before the valve can move back far enough to close the port. This, of course, makes the cut-off take place later in the stroke, and shortens the expansion. It is likewise evident that if the valve had more lap, this extra lap would extend beyond the port when the center of the valve was in the position shown. Therefore, the valve would cut off earlier in the stroke and the expansion would be lengthened. Hence, *increasing the outside lap means an earlier cut-off and an increased expansion, while decreasing the outside lap means a later cut-off and a diminished expansion.*

Considering the inside lap, it is evident from Fig. 252 that, if the inside lap had been less, the exhaust port would not have closed so soon, and, consequently, the compression would have begun later; had the inside lap been greater, the compression would have begun earlier. Fig. 253 shows that with a diminished inside lap, the exhaust (usually termed **release**) would begin earlier, while with an increased inside lap, the release would have taken place later in the stroke. Hence, *increasing the inside lap increases compression and delays the release, while diminishing the inside lap decreases compression and hastens release.*

1245. Lead.—A valve is said to have **lead** when it commences to open the steam port just before the piston

reaches the end of the stroke. The amount of lead is measured by the distance between the edge of the valve and edge of the port from which the valve is traveling. In Fig. 255, the lead is the distance *a b*. Most engineers give their

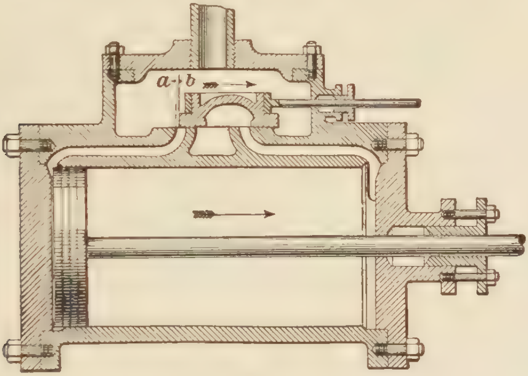


FIG. 255.

valves lead in order to have the clearance space filled with steam at boiler pressure when the piston begins its stroke. The effect of lead on the angular advance of the eccentric is evidently the same as an increase of lap; i. e., it increases the angular advance. Its effect on the distribution of steam will be discussed farther on.

1246. In the preceding pages, where the steam distribution has been discussed, it has been assumed that the engine "runs over." When the engine "runs under," the

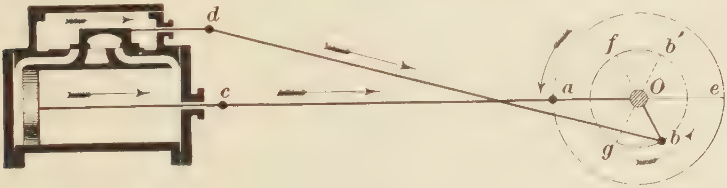


FIG. 256.

steam distribution and the piston and valve movements will be precisely the same as before, but the position of the center of the eccentric relative to the crank-pin will be changed. To determine the position of the eccentric in this case draw

the horizontal diameter $a\ e$ of the crank circle, as shown in Fig. 256. $O\ b'$ represents the position of the eccentric radius when the piston is just beginning the forward stroke and the engine runs over. Draw $b'\ b$ perpendicular to $a\ e$ and through the point b'; it intersects the valve circle in b, and b is the position of the center of the eccentric when the engine runs under and is about to begin the forward stroke. It is easy to see that this is so, for the valve and piston must both have a forward movement at this point, whether the engine runs over or under. If the eccentric radius were placed so as to occupy the position $O\ f$, the forward movement of the piston and downward movement of the crank would cause the valve to move to the left, closing instead of opening the port. It cannot be in the position $O\ g$, for that would throw the valve too far back. $O\ b$ is the only position in which the eccentric radius can be placed to give the valve the same movement when the engine runs under that it would have if placed in the position $O\ b'$ and the engine ran over. In both cases the valve has the same forward movement, while the center of the eccentric is passing from b or b' to the horizontal position $O\ e$.

1247. Rocker Arms.—It frequently happens that the eccentric cannot be so located on the shaft (in the direction of its length) that the eccentric rod and valve stem shall be in the same straight line. It can never be done when the valve is on top of the cylinder without inclining the valve seat, now very seldom done, and, with the valve on the side of the cylinder, other considerations, such as the location of the fly-wheel, may interfere. In such cases as this a lever, or rocker arm, may be employed.

There are four cases which may arise when using a rocker arm: 1. The travel of the valve, the throw of the eccentric, and the direction of motion of the valve and eccentric may be the same as before. 2. The direction of motion of the valve and eccentric may be the same as before, but the travel of the valve may be greater than the throw of the eccentric. 3. The travel of the valve and the throw of the eccentric may

be the same as before, but the eccentric may move in the opposite direction. 4. The travel of the valve may be greater than the throw of the eccentric, and the direction of motion may be opposite that in Cases 1 and 2.

The first case is clearly shown in Fig. 242. It is perfectly evident that when the eccentric rod *11* moves to the left that the valve rod *13* will also move to the left, being compelled to do so by reason of the rocker arm *12*. It is also plain that the amount of horizontal movement of the valve rod will also be the same as it would be if the eccentric were attached directly to the valve rod, thus getting rid of the rocker arm. The reason for using the rocker arm in this case is that the eccentric rod axis and valve stem axis are not in the same straight line, the eccentric being then thrown too far to the left by the main bearing R'. The valve seat could, in this case, have been placed further out from the center of the cylinder, so as to bring the axes of the two rods in line. This, however, would have made the steam and exhaust ports that much longer. Since it is considered to be an advantage to have ports as short as possible, a rocker arm was used.

1248. Sometimes the valve travel is such that if the eccentric were made to have the same throw it would be inconveniently large. In such a case the eccentric and its connection to the rocker arm may be left in the same position as shown in Fig. 242, but the valve and its seat may be

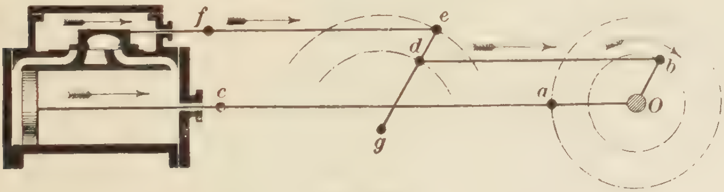


FIG. 257.

raised, the valve stem connecting to the rocker arm at a higher point than shown in Fig. 242. The effect of such an arrangement is illustrated in Fig. 257. The direction of motion of valve and eccentric is the same as in Figs. 247 to

254, but the throw of the eccentric is less than the travel of the valve by the ratio $g\,d : g\,e$; i. e., if the valve travel is 4″ and $g\,d = 12''$ and $g\,e = 15''$, the throw of the eccentric will be $4'' \times \frac{12}{15} = 3.2''$. If the engine runs under, the position of

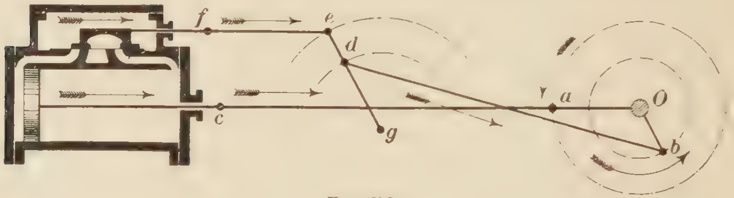


FIG. 258.

the center of the eccentric will be the same as in Fig. 258, and may be found by the same method that was given for finding it in the case shown in Fig. 256.

1249. In all three of the cases just described the direction of motion of the valve and of the eccentric has remained the same as if there had been no rocker arm, and both points of connection *d* and *e* of the valve stem and eccentric rod to the rocker arm were on the *same* side of the pivot *g*. Suppose that the valve had been placed on the top of the cylinder, and it had been found more convenient to place the pivot of the rocker between the connections of the rocker arm to the valve stem and eccentric rod, as shown in Fig. 259; then, when *d* moves to the right, along the dotted arc whose center is at *g*, *e* moves to the left. Consequently, if

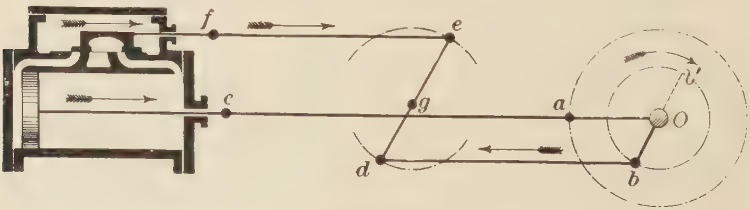


FIG. 259.

the eccentric center was in the position $O\,b'$, and the engine was running in the direction of the arrow, the valve would move backwards instead of ahead. To overcome this difficulty, the eccentric is shifted 180° to the position $O\,b$, then

a movement of *b* in the direction of the arrow will throw *d* to the left and *e* to the right, as is easily seen.

If *g d* and *g e* are equal, the valve travel and the throw of the eccentric will be equal, fulfilling the conditions for Case 3. If *g d* is less than *g e*, the throw of the eccentric will be less than the valve travel by the ratio *g d* : *g e*. For example, suppose that *g e* = 20″ and *g d* = 15″ and the valve travel is 5″; then, the throw of the eccentric will be $5'' \times \frac{15}{20} = 3\frac{3}{4}''$.

1250. Fig. 260 shows the position of the eccentric center when the engine runs under, and the rocker arm is of the same design as the one shown in Fig. 259. If there were no rocker arm, the eccentric center would be at *b′*, as in Fig. 256, but since the rocker arm changes the direction of motion in this case, the eccentric is turned around 180°, to a point diametrically opposite.

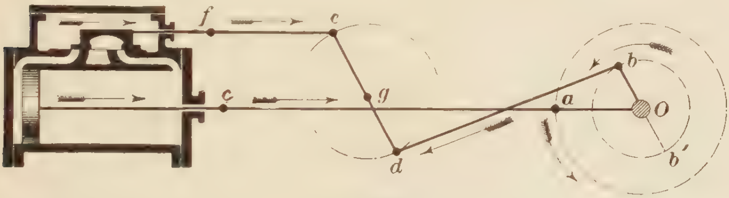


FIG. 260.

The following rule may be applied to any engine whose valves cut off by their outside edges, as has been done in all the previous cases:

Rule.—*Place the crank in the position O a, and the eccentric in the position O b, as shown in Fig. 247, e O b being the angle of advance. If the engine runs over and the rocker arm does not reverse the direction of motion of the eccentric, the eccentric is now correctly set. If the engine runs under, the eccentric should be placed in the position shown in Fig. 256, according to the rule given in connection with that figure. If the engine has a rocker arm whose pivot lies between the point of connection with the valve stem and eccentric rod, and the engine runs over, place the eccentric center diametrically opposite the position shown in Fig. 247. If the engine runs*

under, and the pivot of the rocker arm lies between the two points of connection, place the eccentric center diametrically opposite the position shown in Fig. 256.

The methods of setting the eccentric, when the valve cuts off by its inside edges, and the determination of the proper amount of lap and lead, will be treated of in Arts. **1621** to **1636.**

1251. Dead Centers.—When the piston has reached the end of either stroke, the piston rod, connecting rod, and crank are all in one straight line, and the entire steam pressure on the piston is transmitted directly to the shaft and bearings, none of it being used to turn the crank. This is clearly shown in Fig. 256. It is evident that there is no turning force on the crank due to the steam pressure when the reciprocating parts are in the position shown. When the crank occupies this position, it is said to be on its **dead center.** There are two dead center positions, $O\,a$ and $O\,c$, diametrically opposite each other, corresponding to the two extreme positions of the piston. When the crank occupies the position $O\,a$ it is said to be on its **interior** dead center, and when it occupies the position $O\,c$, on its **exterior** dead center.

1252. Clearance.—When the crank is on a dead center, and the piston at the end of its stroke, there is always a space between the piston and the cylinder head. The volume of the space plus the volume of the one *steam* port leading into it is called the **clearance.** Thus, in Fig. 255 the piston is at the end of its return stroke, and the clearance is the volume of the space between the piston and the left cylinder head, plus the volume of the left steam port. In other words, the clearance may be defined as the volume of steam between the valve and the piston, when the latter is at the end of its stroke. The clearance of an engine may be found by putting the engine on a dead center and pouring in water until the space between the piston and the cylinder head, and the steam port leading into it, is filled. The volume of the water poured in is the clearance.

The clearance may be expressed in cubic feet or cubic inches, but it is more convenient to express it as a percentage of the volume swept through by the piston. For example, suppose the clearance volume of a 12″ × 18″ engine is found to be 128 cu. in. The volume swept through by the piston per stroke is $12^2 \times .7854 \times 18 = 2{,}035.8$ cu. in. Then, the clearance is $\frac{128}{2{,}035.8} = .063 = 6.3\%$. The clearance may be as low as $\frac{1}{2}\%$ in Corliss engines, and as high as 14% in very high speed engines.

1253. Theoretically, there should be no clearance, since the steam which fills the clearance space does no work except during expansion; it is exhausted from the cylinder during the return stroke, and represents so much dead loss. This is remedied, to some extent, by compression. If the compression were carried up to the boiler pressure, there would be very little, if any, loss, since it would then fill the entire clearance space at boiler pressure, and the amount of fresh steam needed would be the volume displaced by the piston up to the point of cut-off, the same as if there were no clearance. It is not practicable to build an engine without any clearance, owing to the formation of water in the cylinder due to the condensation of steam, particularly when starting the engine. As water is practically incompressible, some part of the engine would be broken when the piston reached the end of its stroke, provided there were no clearance space for the water to collect in; usually, the cylinder heads would be blown off. Neither is it practicable to compress to boiler pressure, as a general rule, for that causes too great strains on the engine. Automatic cut-off high-speed engines of the best design, with shaft governors, usually compress to about half the boiler pressure, and have a clearance of from 7% to 14%.

Corliss engines have but very little compression, owing to the construction of the valve; they, likewise, have very little clearance.

1254. Real and Apparent Cut-Off and Ratio of Expansion.—The **apparent cut-off** is the ratio between the portion of the stroke completed by the piston at the point of cut-off, and the total length of stroke. For example, if the length of stroke is 48 inches, and the steam is shut off from the cylinder just as the piston has completed 15 inches of the stroke, the apparent cut-off is $\frac{15}{48} = \frac{5}{16}$.

The **real cut-off** is the ratio between the volume of steam in the cylinder at the point of cut-off, and the volume at the end of the stroke, both volumes including the clearance of the end of the cylinder in question. If the volume of steam in the cylinder (including the clearance) at the point of cut-off is 4 cu. ft., and the volume (including the clearance) at the end of the stroke is 6 cu. ft., the *real* cut-off is $\frac{4}{6} = \frac{2}{3}$.

The **ratio of expansion**, also called the **number of expansions**, is the ratio between the volume of steam (including the steam in the clearance space) at the end of the stroke, and the volume, including the clearance, at the point of cut-off. It is the reciprocal of the real cut-off. For example, if the volume at the end of the stroke is 8 cu. ft., and at cut-off is 5 cu. ft., the ratio of expansion is $\frac{8}{5} = 1.6$; in ordinary language, the steam would be said to have *one and six-tenths expansions.* The corresponding real cut-off would be $\frac{5}{8}$.

Let e = the number of expansions;
i = the clearance, expressed as a per cent. of the stroke;
k = the real cut-off;
k_1 = the apparent cut-off;
r = the apparent number of expansions $= \frac{1}{k_1}$.

$$\text{Then, } e = \frac{1}{k} \text{ and } k = \frac{1}{e}. \qquad \textbf{(96.)}$$

$$k = \frac{k_1 + i}{1 + i}. \qquad \textbf{(97.)}$$

EXAMPLE.—The length of stroke is 36 inches; the steam is cut off when the piston has completed 16 inches of the stroke; the clearance is 4 per cent. Find the apparent cut-off, the real cut-off, and the number of expansions.

SOLUTION.—Apparent cut-off $= \frac{16}{36} = \frac{4}{9} = .444$. Ans.

Real cut-off $= k = \frac{k_1 + i}{1 + i} = \frac{.444 + .04}{1 + .04} = \frac{.484}{1.04} = .465$. Ans.

Number of expansions $= e = \frac{1}{k} = \frac{1}{.465} = 2.15$. Ans.

EXAMPLES FOR PRACTICE.

1. Length of stroke, 18″; apparent cut-off, .4; clearance, 7.5%. Find (*a*) real cut-off, and (*b*) real number of expansions.

Ans. { (*a*) .442. (*b*) 2.262.

2. Length of stroke, 66″; clearance, 4%; steam cuts off at 14½″. Find (*a*) real and (*b*) apparent cut-off in per cent. of stroke, and (*c*) real and (*d*) apparent number of expansions.

Ans. { (*a*) 24.97%. (*b*) 21.97%. (*c*) 4, nearly. (*d*) 4.552 nearly.

INDICATORS AND INDICATOR CARDS.

1255. The arrangement described in connection with Figs. 247 to 254, for recording the steam pressure at all points of the stroke of the piston, would be impossible to put into actual operation. Again, the diagram traced by the pencil would be altogether too large to be handled conveniently. To overcome these objections, an instrument called the **indicator** is used. The principal reason for obtaining a diagram of this kind is that it affords a ready means of computing the mean pressure of the steam upon the piston during one stroke. If the mean pressure on both sides of the piston, the length of the stroke, and the number of strokes per minute are known, the horsepower of the engine can be easily found.

1256. Fig. 261 shows the general appearance of an indicator, and Fig. 262 shows one in section. The instrument consists essentially of a cylinder *a* containing the piston *g* and spiral spring *d*. By turning a cock connected

to the small pipe to which the indicator is attached, steam may be admitted to, or shut off from, the cylinder of the indicator at pleasure. When steam is admitted through the channel *s*, Fig. 262, its pressure causes the piston *g* to rise. The spiral spring *d* is compressed, and resists the upward movement of the piston. The height to which the piston rises should then be in exact proportion to the pressure of the steam, and, as the steam pressure rises and falls, the piston must rise and fall accordingly.

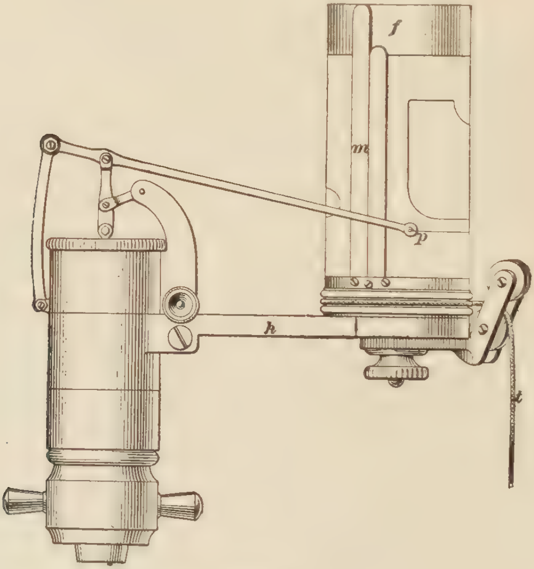


FIG. 261.

To register this pressure, a pencil might simply be attached to the end of the piston rod *c*, the point of the pencil being made to press against a piece of paper. It is desirable, however, to restrict the maximum travel of the piston to about half an inch; while the height of the card may advantageously be two inches or more. To obtain a long range of the pencil, combined with a short travel of the piston, the pencil is attached at *p* to the long end of the lever *n k p*. The fulcrum of the lever is at *n*. The piston

rod is connected to it at k through the link i. The pencil motion is thus $\frac{pn}{nk}$ times the piston travel. This ratio $\frac{pn}{nk}$ is, for most indicators, either 4, 5, or 6. The point p is forced

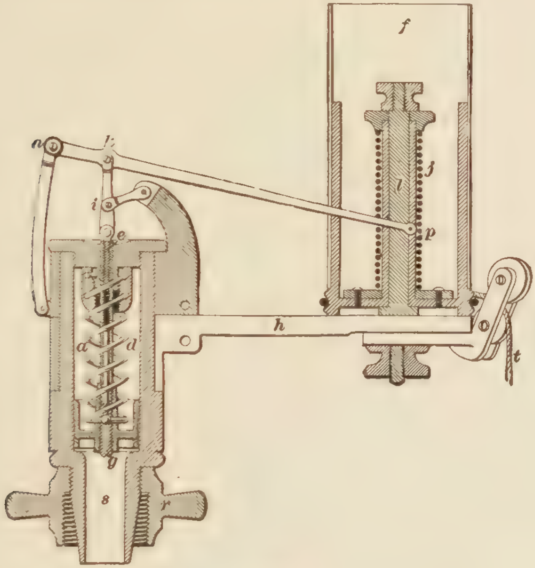


FIG. 262.

to move in a vertical straight line by the arrangement of the links and joints i, e, n, k.

1257. The height to which the piston will rise under a given steam pressure depends upon the stiffness of the spring. Indicators are usually furnished with a number of springs of varying degrees of stiffness, which are distinguished by the numbers 20, 30, 40, etc.

These numbers indicate the pressure per square inch required to raise the pencil one inch. Thus, if a 40 spring is used, a pressure of 40 pounds per square inch raises the pencil one inch, and, therefore, the vertical scale of the diagram is 40 pounds per inch. That is, the vertical distance in inches of any point on the diagram from the atmospheric line, multiplied by 40, gives the gauge pressure per square

inch at that point. The scale of the spring chosen should not be less than half the boiler pressure. For example, we would choose a 40 spring for a steam pressure of 75 pounds per square inch.

1258. The indicator, however, must not only register pressures, but it must register them in relation to the position of the piston. To accomplish this object, a cylindrical drum *f* is provided. This drum can be revolved on its axis *l*, by pulling the cord *t*, which is coiled around it. When the pull is released, the spring *j* rotates the drum back to its original position. If now the cord *t* be attached to some part of the engine which has a motion proportional to the motion of the piston, the motion of the drum will also be proportional to the motion of the piston.

1259. To attach the indicator to the engine, a hole is drilled in the clearance space of the cylinder and tapped for

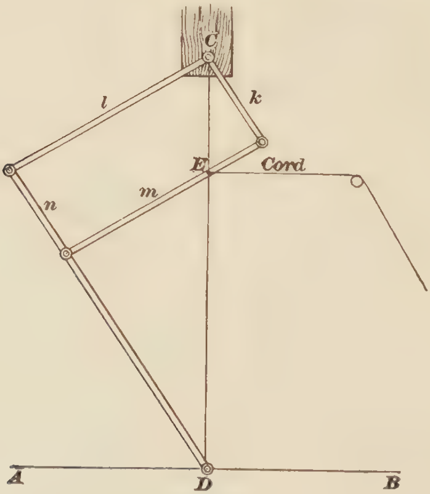


FIG. 263.

a ½" nipple. The nipple has an elbow, into which is screwed the indicator cock. The indicator is then attached directly to the cock by the nut *r*, the conical projection *s* of the indicator wedging tightly into the cock to prevent the

leakage of steam. It is preferable to have an indicator at each end of the cylinder, but if that is not convenient, one indicator may be connected with both ends of the cylinder by means of a three-way cock, as shown in Fig. 264.

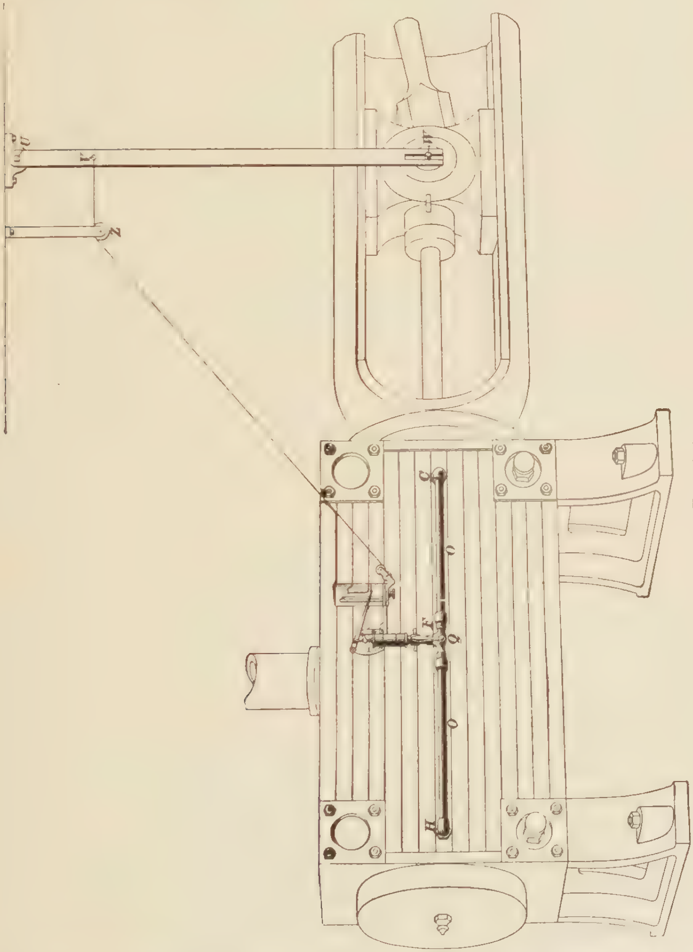


Fig. 264.

The motion of the drum cord is usually obtained from the cross-head. Since the stroke of the engine is nearly always greater than the circumference of the drum, the cord cannot be attached directly to the cross-head, and an

arrangement called a **reducing motion** is resorted to. A common form of reducing motion is shown in Fig. 263. It consists of four links joined together in the form of a parallelogram. One of the links n is prolonged, and is pivoted at the end to the cross-head D. The opposite corner of the parallelogram is pivoted to the fixed point C. The cord is attached to the point E on the link m, which point must be on the straight line connecting C and D. $A\,B$ represents the length of the stroke. Letting L represent the length of the indicator diagram, we have the following proportions: $A\,B : L = C\,D : C\,E$, or $\frac{A\,B}{L} = \frac{C\,D}{C\,E}$. This reducing motion is called the pantograph.

1260. Another reducing motion is shown in the "slotted swinging lever," shown in Fig. 264. A pin in the cross-head moves in a slot in one end of the lever, the other end of which is pivoted, at U, to some stationary body. The cord is attached at V. As in the former arrangement,

$$\frac{\text{length of diagram}}{\text{length of stroke}} = \frac{U\,V}{U\,W}, \text{ or}$$

$$U\,V = \frac{\text{length of diagram} \times U\,W}{\text{length of stroke}}.$$

Therefore, to find the distance $U\,V$ between the pivot U and the point V where the cord is attached, multiply the length of the lever by the desired length of diagram, and divide the product by the length of stroke, all the dimensions being taken in inches.

EXAMPLE.—The stroke of an engine is 28 inches; the length of the slotted lever is 6 feet. How far from the pivot must the string be attached to give a diagram $3\frac{1}{2}$ inches long?

SOLUTION.— $\frac{72 \times 3\frac{1}{2}}{28} = 9$ inches.

TO TAKE THE CARD.

1261. The instrument being attached to the engine, as explained above, slip a blank card over the drum, as shown in Fig. 261. Fasten the cord to the reducing gear, taking care that it is not too short, and that the drum does not

"strike" at either end of the stroke. (For convenience, the cord should be divided, one part being attached to the reducing gear, and the other to the drum. One end is provided with a loop and the other with a hook, so that they may be connected and disconnected easily.) Open the cock and let the engine make a few strokes to warm up the indicator; then press the pencil gently against the rotating drum during one revolution. Shut the cock and again press the pencil against the drum, to obtain the atmospheric line. Now, disconnect the drum cord and take off the card.

If but one indicator and a three-way cock are used, as shown in Fig. 264, open the cock to admit steam from one end of the cylinder, and take the card from that end; then turn the cock to admit steam from the other end, and take that card; then shut off the steam entirely, and take the atmospheric line.

1262. Figs. 265 and 266 are cards taken from the head

FIG. 265.

and crank ends of the cylinder, respectively. The different phases during the stroke are very clearly shown.

Thus: *1* is the beginning of the stroke;
2 is the point of cut-off;
3 is the point of release;
4 is the end of the stroke;
5 is the point of compression;
6 is the point of admission.

The lines included between any two of these points have received special names, which are as follows:

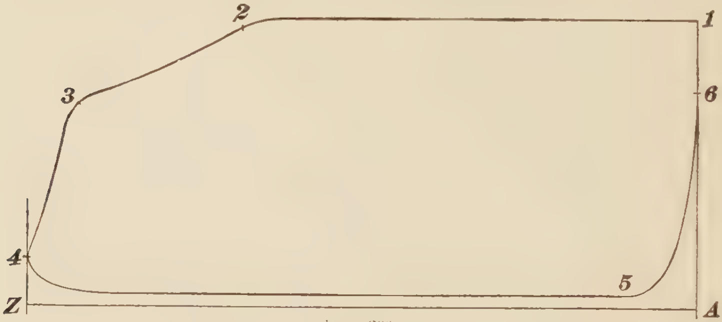


FIG. 266.

6-1 is the admission line;
1-2 is the steam line;
2-3 is the expansion curve;
3-4-5 is the period of release;
4-5 is the back pressure line;
5-6 is the compression curve;
A Z is the atmospheric line.

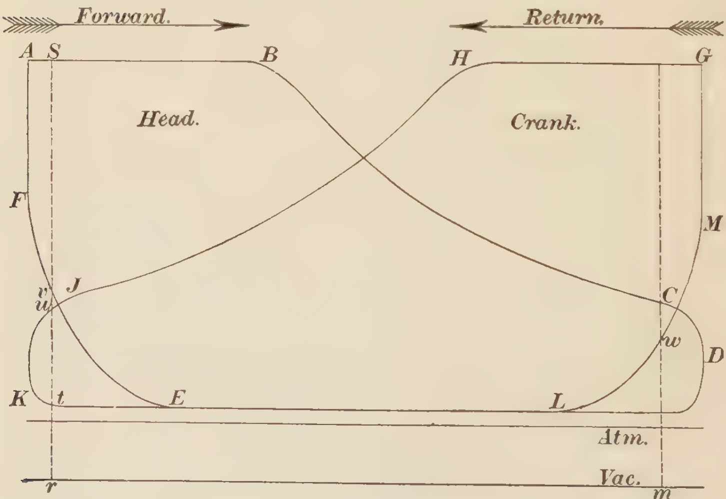


FIG. 267.

1263. If but one indicator is used, the two diagrams may be taken on the same blank as shown in Fig 267.

With the diagrams placed one over the other, as shown, it is very easy to tell exactly what is taking place in the cylinder at any point of the stroke. On the forward stroke the pencil of the indicator describes the line $A\ B\ C\ D$ of the head diagram, if the cock is opened to the head end, or it describes the line $K\ L\ M$, if the cock is opened to the crank end. Likewise, the lines $G\ H\ J\ K$ and $D\ E\ F$ are described during the return stroke.

Suppose the piston is at a position corresponding to r on the forward stroke; the pressure (absolute) urging the piston forward is $r\ S$, while the pressure resisting is $r\ t$. Hence, the net pressure on the piston is $S\ t$. Suppose, now, that the piston is at r on the return stroke; the pressure at the right urging the piston on is $r\ u$, while the pressure on the left is $r\ v$. The net pressure is, therefore, $u\ v$, and is negative; or, in other words, the resistance is greater than the effort.

1264. A double diagram of this character tells at a glance what is taking place at either end of the cylinder at any point of the stroke. Thus, when the piston is on the forward stroke, in the position corresponding to m, the steam in the head end is at the point of release, as shown at C. Draw a line through m perpendicular to the vacuum line. C lies on $A\ B\ C$, and since $K\ L\ M$ is described at the same time as $A\ B\ C$, the intersection of the line through C with the line $K\ L\ M$ is the point corresponding to C. Since w is on the compression line, compression is taking place in the crank end when release occurs in the head end.

HORSEPOWER.

1265. In order to find from the diagram the horsepower exerted by the engine, we must first find the *mean effective pressure.*

The **mean effective pressure,** or M. E. P., is defined as the average pressure urging the piston forward during its entire stroke in one direction, less the pressure that resists its progress.

The mean effective pressure may be found in two ways: 1. The area of the diagram in square inches may be found

by an instrument called the planimeter; the M. E. P. is then found by dividing the area of the diagram in square inches by the length of the diagram in inches, and multiplying by the scale of the spring.

EXAMPLE.—The area of the diagram is 4.2 sq. in., and the length is 3.5 in.; a 40 spring being used, find the M. E. P.

SOLUTION.—$\frac{4.2}{3.5} \times 40 = 48$ lb. per sq. in. M. E. P. Ans.

2. Where a planimeter is not available, the following method of finding the M. E. P. is fairly rapid and accurate: Draw tangents to each end of the diagram perpendicular to the atmospheric line. Divide the horizontal distance between the tangents into 10 or more equal parts (10 or 20 parts are the most convenient, but any other number may be used). Indicate by a dot on the card the center of each division, and draw lines through these dots parallel to the tangents from the upper line to the lower line of the card. On a strip of paper mark off successively the length of these lines, the total length thus representing the sum of all the lines. Divide this total length by the number of the lines used, and multiply the quotient by the scale of the spring. The result will be the M. E. P. This method is the same as that given in Art. 1159.

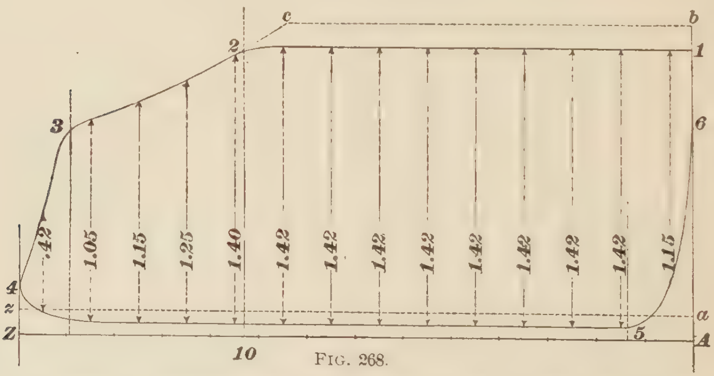


FIG. 268.

EXAMPLE.—The projection of the head end card of Fig. 265 (see Fig. 268) upon the atmospheric line is the distance $A Z$, and it is divided, in this case, into 14 equal spaces. The length of each of the perpendicular lines drawn through the card opposite the centers of these spaces, is

marked on the lines themselves, and the sum of these lengths is 18.11 inches. The scale of the spring used in obtaining the card was 40 pounds; therefore,

$$\frac{18.11}{14} \times 40 = 51.74 \text{ pounds per square inch} =$$

the M. E. P. of the "head" end card.

EXAMPLE.—The projection of the crank card of Fig. 266 (see Fig. 269), upon the atmospheric line, is the distance $A\ Z$, and it is divided in this case, into 14 equal spaces. The length of each of the perpendicular lines drawn through the card opposite the centers of these spaces, is marked on the lines themselves, and the sum of these lengths is 17.78 inches. The scale of the spring is 40 pounds; therefore,

$$\frac{17.78}{14} \times 40 = 50.8 \text{ pounds per square inch} =$$

the M. E. P. of the "crank" end card.

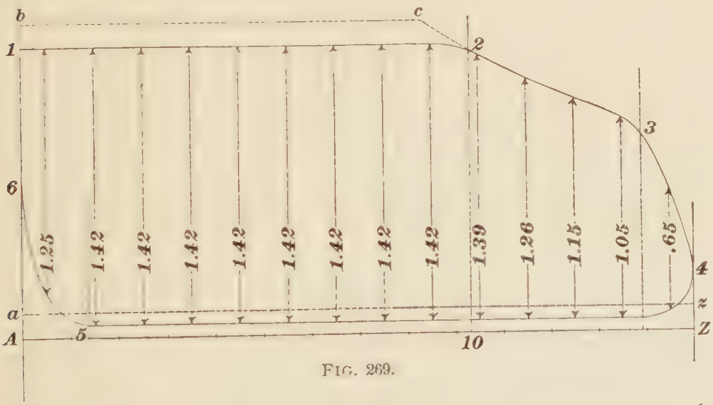


FIG. 269.

Therefore, the M. E. P. in the cylinder during a complete revolution of the crank is

$$\frac{51.74 + 50.8}{2} = 51.27 \text{ pounds per square inch.}$$

1266. The reason for dividing the diagram into 10 parts instead of some other number, is that it shortens the work of calculation. Thus, in the two examples just given, if the number of divisors had been 10 instead of 14, and the sum of the ordinates had been 12.94″, the mean ordinate would have been $\frac{12.94}{10} = 1.294$, and the M. E P., $1.294 \times 40 = 51.76$ lb. per sq. in. All that is necessary is to add the

ordinates and shift the decimal point one place to the left to obtain the mean ordinate when the diagram is divided into 10 equal parts. This method saves the time required to divide by some inconvenient number, as 14.

1267. Sometimes the expansion line of the card will fall below the back pressure line, as shown in Fig. 270. In such a

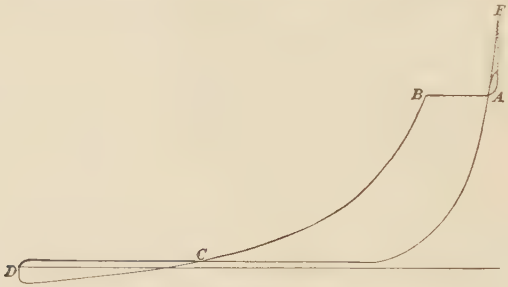


FIG. 270.

case the area of the loop *C D* must be subtracted from the remainder of the card *A B C*. When the planimeter is used, the subtraction is made automatically by the instrument; but when the card is divided into parts by the method of ordinates the sum of the ordinates of *C D* must be subtracted from the sum of those of *A B C A*. The result divided by the number of spaces will give the mean ordinate; multiplying this by the scale of the spring will give the M. E. P., as before.

1268. We have now all the materials for finding the work done in the engine cylinder, expressed in horsepowers.

Work is the product of force into the distance through which the force moves. In the case of the engine cylinder the total force is the M. E. P. per square inch multiplied by the area of the piston; and the distance moved through in a minute is the number of strokes in a given time multiplied by the length of stroke.

Let P represent the M. E. P. in lb. per sq. in.;
A represent the area of piston in sq. in.;
L represent the length of stroke in ft.;
N represent the number of strokes per min.

Then, the work done per minute is $P\,L\,A\,N$ ft.-lb.

One horsepower = 33,000 ft.-lb. per min.

Therefore, the indicated horsepower of the engine is found from the formula,

$$\text{I. H. P.} = \frac{P\,L\,A\,N}{33{,}000}. \qquad \textbf{(98.)}$$

1269. When the point of real cut-off and the steam pressure at the beginning of the stroke are known, the M. E. P. may be found approximately by the following formula:

$$\text{M. E. P.} = \frac{.9\,P(1 + 2.3 \log e)}{e} - .9\,p, \qquad \textbf{(99.)}$$

in which P = absolute steam pressure = gauge pressure + 14.7 pounds;

e = ratio of expansion;

p = absolute back pressure.

p is usually taken as about 3 pounds for condensing engines, and 17 pounds for non-condensing engines.

Example.—A non-condensing engine cuts off at $\frac{2}{3}$ stroke. The clearance is 5%; the gauge pressure is 59.3 pounds. What is the approximate M. E. P.?

Solution.—$k_1 = \frac{2}{3}$. $k = \frac{k_1 + i}{1 + i} = \frac{.66\frac{2}{3} + .05}{1.00 + .05} = \frac{.71\frac{2}{3}}{1.05} = \frac{215}{315} = \frac{43}{63}$. $e = \frac{1}{k} = \frac{63}{43}$, $\log \frac{63}{43} = .16587$. $P = 59.3 + 14.7 = 74$ lb. M. E. P. = $\frac{.9\,P(1 + 2.3 \log e)}{e} - .9\,p = \frac{.9 \times 74 \times (1 + 2.3 \times .16587)}{\frac{63}{43}} - .9 \times 17 =$ 47.5 lb.

Example.—The diameter of the piston of an engine is 10 inches, and the length of stroke 15 inches; it makes 250 revolutions per minute, with a M. E. P. of 40 pounds per square inch; what is the horsepower?

Solution.—As it is not stated whether the engine is single or double acting, assume that it is double-acting. Then, the number of strokes is $250 \times 2 = 500$ per minute. Substituting in formula **98**,

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{40 \times \frac{15}{12} \times (10^2 \times .7854) \times 500}{33{,}000} = 59.5 \text{ H. P.}$$

1270. The product $L\,N$ of the above formula gives the total distance in feet traveled by the piston per minute. It is

called the **piston speed.** If the length of stroke L be taken in inches, the piston speed becomes $\frac{LN}{12}$. If R = number of revolutions per minute, $\frac{LN}{12} = \frac{L \times 2R}{12} = \frac{LR}{6}$.

Letting S represent the piston speed in feet per minute,

$$\left.\begin{aligned} S &= \frac{LR}{6} \\ L &= \frac{6\,S}{R} \\ R &= \frac{6\,S}{L} \end{aligned}\right\} \quad \textbf{(100.)}$$

The piston speeds used in modern practice are about as follows:

	ft. per min.
Small stationary engines, -	300 to 600.
Large stationary engines,	600 to 1,000.
Corliss engines, - - - -	400 to 750.
Locomotive engines, - -	600 to 1,200.

1271. Having given the I. H. P. of the engine and knowing the available M. E. P., there are two methods of calculating the length of stroke and diameter of piston:

1. We may assume the number of revolutions, and the ratio of the length of stroke to the diameter of cylinder.

2. We may assume a suitable piston speed, and choose the number of revolutions and length of stroke to correspond.

An example will serve to illustrate the above methods.

Given an engine which is to develop 250 I. H. P. with a M. E. P. of 50 pounds per square inch. Find the diameter of piston and length of stroke.

First, let us assume that the engine makes a certain number of revolutions per minute, as 75, and that the length of stroke in *inches* is say twice the diameter of the piston. Substituting in formula **98,**

$$250 = \frac{50 \times L \times A \times (75 \times 2)}{33{,}000},$$

$$\text{or } LA = \frac{33{,}000 \times 250}{50 \times 75 \times 2} = 1{,}100.$$

In the above expression, L is taken in *feet.* It is more convenient to use inches, so we will multiply both sides of the equation by 12.

$$12\ L_{ft.} \times A = L_{in.} \times A = 13{,}200.$$

But $A = .7854\ D^2$ and $L = 2\ D$, according to the above assumption.

Substituting, $L\ A = 2\ D \times .7854\ D^2 = 1.5708\ D^3 = 13{,}200.$

$$D^3 = 8{,}408.$$

$$D = \sqrt[3]{8{,}408} = 20.33 \text{ in.}$$

$$L = 20.33 \times 2 = 40.66 \text{ in.}$$

Second method. We may assume a certain piston speed, say 500 feet per minute; then, as above,

$$250 = \frac{50 \times L\ A\ N}{33{,}000}.$$

But the piston speed $= 500$ feet $= L\ N.$

$$\text{Therefore, } 250 = \frac{50 \times A \times 500}{33{,}000},$$

$$\text{or } A = \frac{250 \times 33{,}000}{50 \times 500} = 330 \text{ sq. in.}$$

$$.7854\ D^2 = 330$$

$$D^2 = 420$$

$$D = 20\tfrac{1}{2} \text{ inches.}$$

If we assume, as before, 75 revolutions per minute, the length of the stroke, from formula **100,** is

$$\frac{500 \times 6}{75} = 40 \text{ inches.}$$

EXAMPLES FOR PRACTICE.

1. The mean ordinates of two diagrams taken from each end of the cylinder of an $18'' \times 20''$ non-condensing engine running at 200 R. P. M. (revolutions per minute) are respectively .72″ and .76″ long. The scale of spring being 80, what is the horsepower of the engine?

Ans. 304.335 H. P.

2. In the above engine, assume the initial pressure to be 118 lb. per sq. in., gauge; the apparent cut-off as $\frac{1}{4}$, and the clearance as 8%. (*a*) Find the theoretical M. E. P., and (*b*) the horsepower.

Ans. $\begin{cases} (a)\ 64.41 \text{ lb. per sq. in.} \\ (b)\ 331.12 \text{ H. P.} \end{cases}$

3. An engine running at 165 R. P. M. has a stroke of 28″; what is the piston speed? Ans. 770 ft. per min.

4. If an engine has a piston speed of 960 ft. per min., and runs at 72 R. P. M., what is the length of the stroke? Ans. 80 in.

5. Initial pressure, 82 lb., gauge; number of expansions, 1.83; back pressure, 4.2 lb., absolute; what is the theoretical M. E. P.?
Ans. 72.484 lb. per sq. in.

6. I. H. P., 536.42; piston speed, 480 ft. per min.; M. E. P., 61.15 lb. per sq. in. Find diameter of cylinder to nearest $\frac{1}{8}''$. Ans. $27\frac{3}{4}$ in.

7. A 16″ × 20″ engine develops 138 I. H. P. with 35 lb. M. E. P.; how many R. P. M. does it make? Ans. 194.2 R. P. M.

8. A 54″ × 66″ Porter-Allen non-condensing engine develops 1,382.4 I. H. P., with an initial pressure of 63 lb., gauge, when cutting off at $\frac{1}{5}$ stroke and running at 82 R. P. M. (*a*) What is the M. E. P.? (*b*) With a back pressure of 1 lb. above the atmosphere and a clearance of 3%, what would be the theoretical I. H. P. calculated by formulas **98** and **99**?
Ans. $\begin{cases} (a) \text{ 22.083 lb. per sq. in.} \\ (b) \text{ 1,558.1 I. H. P.} \end{cases}$

9. A 16″ × 14″ engine runs at 240 R. P. M.; what is the piston speed? Ans. 560 ft. per min.

10. If the average M. E. P. of the engine in the last example is 41.73 lb. per sq. in. and the diameter of the piston rod is 4″, what is the I. H. P., taking the piston rod into consideration? Ans. 137.932 H. P.

1272. From the measurement of the indicator diagrams has been obtained what we have termed the indicated horsepower, or I. H. P.—that is, the total horsepower developed in the engine cylinder. One portion of the I. H. P. is absorbed in overcoming the friction of the engine itself. The remainder is available for doing the required work.

The power absorbed by the engine itself is termed **friction horsepower.**

The power available for doing useful work is termed the **net** or **actual horsepower.**

1273. The actual horsepower of any engine is found by first computing its I. H. P. from a set of indicator cards taken when the engine is running under full load, and then subtracting from this the I. H. P. computed from a set of indicator cards taken when the engine is running under no load, but making the same number of revolutions per minute as

above. The horsepower developed by the engine in this last case will only be sufficient to keep the working parts of the engine in motion at the same speed. To produce this result, some means will have to be resorted to of checking the steam supply. These will be discussed later.

EXAMPLE.—Indicator cards taken from an engine when running under full load, and having a piston speed of 498 feet per minute, showed an *indicated horsepower* of 242.7. With the same piston speed, and running under no load, the indicator cards showed an *indicated horsepower* of 75.2. Therefore, $242.7 - 75.2 = 167.5$, which is the *actual horsepower* of the engine.

1274. The **mechanical efficiency** of an engine is the ratio of the *actual horsepower* to the *indicated horsepower;* or it is the per cent. of the mechanical energy developed in the cylinder which is utilized in the doing of useful work.

To find the efficiency of an engine, when the *indicated* and *actual horsepowers* are known :

Rule.—Divide the *actual horsepower* by the *indicated horsepower.*

EXAMPLE.—The indicated horsepower of an engine is 242.7, and the actual horsepower is 197.5. Therefore, $\frac{197.5}{242.7} = 81.38$ per cent. efficiency.

The mechanical efficiency of engines in good order varies from 75 to 90 per cent.

1275. The **thermal efficiency** of the engine is the same as that of any other heat engine. This was shown in Art. **1182** to be $\frac{T_1 - T_2}{T_1}$, where T_1 is the absolute temperature of the entering steam, and T_2 the absolute temperature of the exhaust steam.

EXAMPLE.—The pressure of the entering steam is 100 pounds above vacuum, and at exhaust it is 16 pounds above vacuum; what is the thermal efficiency ?

SOLUTION.—

Temperature of incoming steam, from table, 327.625°.
" " exhaust " " " 216.347°

Absolute $T_1 = 327.625 + 460 = 787.625$.
" $T_2 = 216.347 + 460 = 676.347$.

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1} = \frac{787.625 - 676.347}{787.625} = 14.13 \text{ per cent.}$$

READING INDICATOR DIAGRAMS.

1276. The determination of the I. H. P. is not the only or the most important function of the indicator. By its use, defects in steam distribution may be detected, the correction of which may result in largely increased economy in the working of the engine.

The form of a good diagram depends largely upon the type of engine, style of valve, and speed. The same style of diagram is not possible or desirable from all engines.

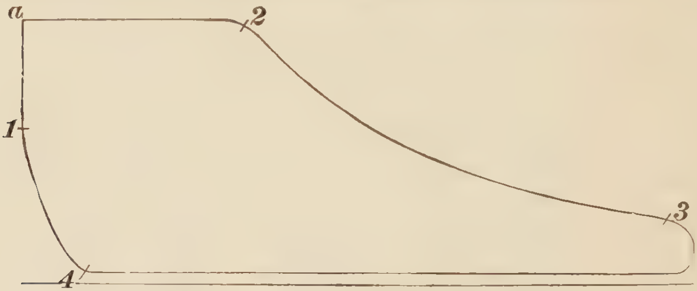


FIG. 271.

Some of the most common faults in steam distribution are given below :

In Fig. 271, *1* is the admission.
2 " " cut-off.
3 " " release.
4 " " compression.

I.	Admission	may	be	too	early.
II.	"	"	"	"	late.
III.	Cut-off	"	"	"	early.
IV.	" "	"	"	"	late.
V.	Release	"	"	"	early.
VI.	"	"	"	"	late.
VII.	Compression	"	"	"	early.
VIII.	"	"	"	"	late.

1277. Case I.—The effect upon the diagram of a too early admission is shown in Fig. 272. It is seen that the admission line *1a* instead of being straight and perpendicular to the atmospheric line, as in Fig. 271, curves

backwards. With a single valve, like the one previously described, all of the other events, cut-off, release and

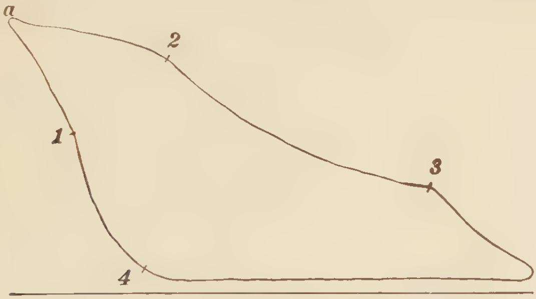


FIG. 272.

compression, are also too early. The remedy is to decrease the angular advance of the eccentric.

1278. Case II.—In this case the admission is too late, and the admission line *1 a*, on a diagram, will curve forwards, as shown in Fig. 273. The remedy is to increase the

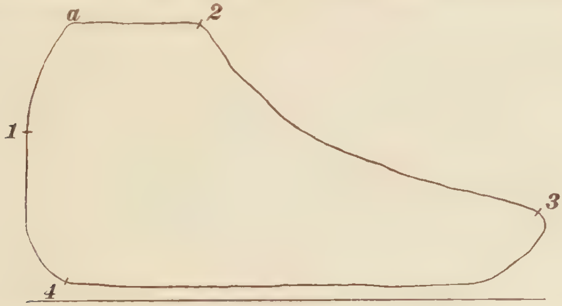


FIG. 273.

angular advance until the admission line *1 a* becomes perpendicular to the atmospheric line. It will be noticed that, in the case of a too late admission, the other events at *2*, and particularly *3* and *4*, are also too late.

1279. Case III.—Cut-off too early. See Fig. 270. Here the steam expands below the back-pressure line and forms a loop. This makes the compression too early also. The figure shown is drawn as if there were no inside lap. With lead, the curve will form a loop, as shown at *A*; without lead, the compression line will extend to *F*. The remedy

is to reduce the amount of outside lap. When computing the M. E. P., the areas of both loops must be subtracted from the area *A B C A*.

1280. Case IV.—Cut-off too late. See Fig. 274. Here it will be noticed that the terminal pressure is very high.

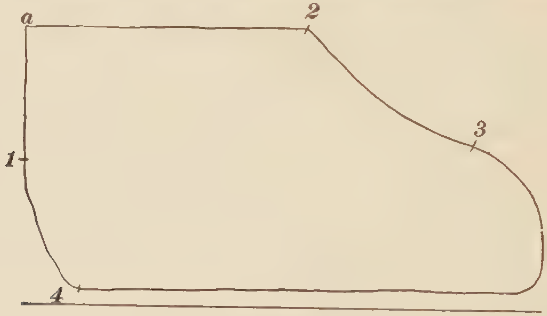


FIG. 274.

When this is the case, a great deal of the benefit of expansion is lost, with its consequent waste of steam.

Rule for Cases III and IV.—Make the cards alike for both ends of the cylinder. For a too early cut-off, lower the boiler pressure or decrease the number of revolutions per minute. For a too late cut-off, raise the boiler pressure or increase the number of revolutions per minute. The cut-off is most correctly equalized by making the terminal pressure at both ends of the cylinder the same.

Case V.—See Fig. 272.

Case VI.—See Fig. 273.

Rule for Cases V and VI.—Arrange the valves so that one-half of the fall of pressure occurs before the piston starts back on the return stroke.

1281. Case VII.—Compression too early.

Fig. 275 shows the effects of too early compression. A loop is formed, as shown in Fig. 270. The area of this loop must be subtracted from the larger area in computing the M. E. P. With the same cut-off and the proper amount of compression, the area gained would be *a b 4 a*, included between the line *4 a* and the dotted line *a b 4*, plus the area of

the loop. The remedy in this case is to decrease the amount of inside lap. The required amount of compression depends

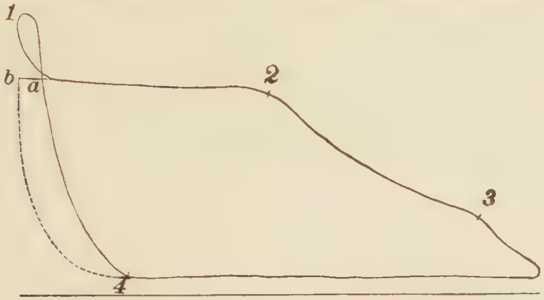


FIG. 275.

upon the speed of the engine, slow-running engines not requiring so much compression as the high-speed engines. In any case, the compression should not extend above the initial or boiler pressure.

It is good practice to compress to about $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines.

All of the above faults are due to valve setting, and can be detected as soon as the indicator is applied.

1282. With a plain slide-valve it will be found that if one of the events of the stroke is early or late, the others

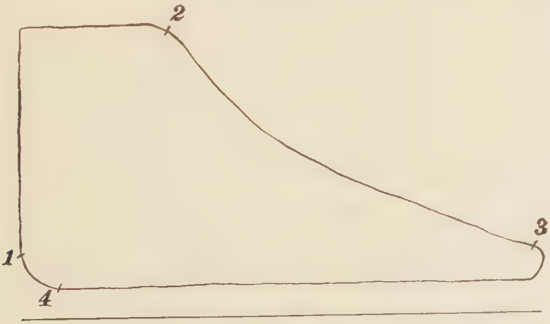


FIG. 276.

are liable to be so also; for example, an early admission usually produces an early release and compression.

When the steam line falls abruptly, as shown in Fig. 272, it may be inferred that the steam is throttled; i. e., either the steam pipe or the port is too small for the required duty. A very high piston speed would also produce this effect.

The card shown in Fig. 276 indicates that the back pressure is excessive. This may be the case when the exhaust port is too small or when the exhaust steam is used for heating purposes, and, in consequence, has to be pushed through coils of pipe.

STEAM CONSUMPTION.

1283. The indicator card also enables us to find approximately the amount of steam consumed by the engine. In referring to the steam consumption, it is customary to take as a unit the *steam consumed per horsepower per hour.*

Take a point a on the expansion line before the release (see Fig. 277); measure the pressure from the vacuum line, and from column 6 of the steam table find the weight

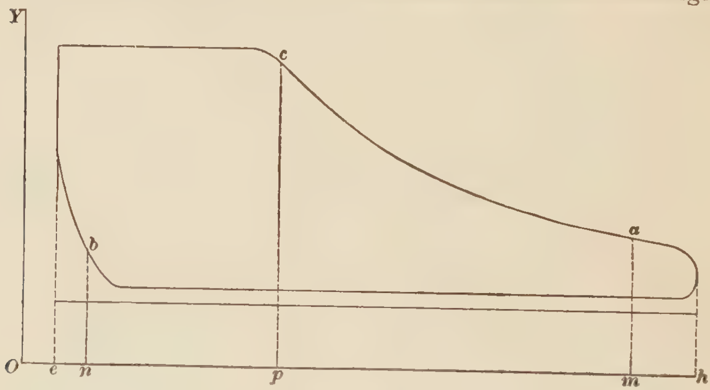


FIG. 277.

of a cubic foot at that pressure. The cubic contents of the cylinder (including the clearance) up to the point a, multiplied by the weight per cu. ft., must give the weight of steam in the cylinder at this instant. Were it not for compression and cylinder condensation, the above weight would represent the steam consumed per stroke. On account of compression, some steam is saved by the early

closure of the exhaust port. To find its weight, take a point b on the compression curve, measure its pressure from vacuum, as before, and compute the weight of the steam in the cylinder up to b. Subtract this from the weight first obtained, and the difference will be the weight of steam per stroke, accounted for by the indicator. Multiply this weight per stroke by the number of strokes per hour and divide by the I. H. P. of the engine. The result will be the steam used per I. H. P. per hour.

EXAMPLE.—Fig. 277 represents an indicator diagram taken from an engine with an 18″ × 24″ cylinder, running at 120 revolutions and developing 130 horsepower. The clearance is 5%. Find the steam consumption per I. H. P. per hour.

SOLUTION.—Project the two ends of the diagram perpendicularly upon the vacuum line, as at e and h; $e\,h$ is then the length of the diagram. Lay off $e\,O$ equal to the clearance—that is, equal to 5% of $e\,h$. Draw $O\,Y$ perpendicular to $O\,h$. Take the point a, near the point of release, and measure the distances $a\,m$ and $O\,m$. Take the point b, somewhere on the compression line, and measure the distances $b\,n$ and $O\,n$. The measurements are found to be:

$$a\,m = 0.71 \text{ inches};$$
$$O\,m = 3.17 \text{ inches};$$
$$b\,n = 0.6 \text{ inches};$$
$$O\,n = \tfrac{1}{3} \text{ inch}.$$

The length of the diagram $= e\,h = 3\frac{1}{3}$ inches; the length of the stroke is 2 feet. Hence, each inch of the length of the card equals $\frac{2}{3\frac{1}{3}}$ $= .6$ foot of stroke. The scale of the indicator spring is 45. Hence, the above measurements reduced to pressures in pounds per square inch and feet of stroke become:

$$a\,m = .71 \times 45 = 31.95 \text{ pounds};$$
$$b\,n = .6 \times 45 = 27 \text{ pounds};$$
$$O\,m = 3.17 \times .6 = 1.9 \text{ feet};$$
$$O\,n = \tfrac{1}{3} \times .6 = .2 \text{ foot}.$$

The area of the piston is $18^2 \times .7854 = 254.47$ sq. in. $= \frac{254.47}{144} = 1.767$ sq. ft. Consequently, the volume of steam in the cylinder, when the piston is at the point represented by a, is $1.9 \times 1.767 = 3.3573$ cu. ft. The volume, when the piston is at b, is $.2 \times 1.767 = .3534$ cu. ft. The weight of a cubic foot of steam at an absolute pressure of 31.95 pounds per sq. in. is found from the steam table to be .078723 pound; and at a

pressure of 27 pounds the weight is .067207 pound. Hence, the weight of the steam in the cylinder is $.078723 \times 3.3573 = .264297$ pound; while the weight of steam saved by compression is $.067207 \times .3534 = .023751$ pound. The steam used per stroke is, therefore, $.264297 - .023751 = .240546$ pound, and the amount used per I. H. P. per hour is

$$\frac{.240546 \times 120 \times 2 \times 60}{130} = 26.645 \text{ pounds.}$$

Suppose the weight of the steam in the cylinder to be calculated by taking the point c, near the point of cut-off. $c p = 1.59$ inches $= 1.59 \times 45 = 71.55$ pounds; $O p = 1\frac{1}{3}$ inches $= \frac{4}{3} \times .6 = .8$ foot of stroke. The volume of steam in the cylinder when the piston is at c is, therefore, $.8 \times 1.767 = 1.4136$ cubic feet. One cubic foot of steam at the pressure of 71.55 pounds, absolute, weighs .168009 pound. The weight of the steam in the cylinder is, therefore, $.168009 \times 1.4136 = .237498$ pound. Subtracting the steam saved by compression, the steam used per stroke is $.237498 - .023751 = .213747$ pound, and the steam per I. H. P. per hour is

$$\frac{.213747 \times 120 \times 2 \times 60}{130} = 23.677 \text{ pounds}$$

Now, unless the valve leaks, the weight of the steam when the piston is at a can be no greater than when it is at c, since no fresh steam has been allowed to enter; but the calculation shows that there is .264297 pound in the cylinder when the piston is at a, and only .237498 pound when the piston is at c. This shows that $.264297 - .237498 = .026799$ pound has been condensed to water by the time the piston has arrived at c, but has been re-evaporated before the piston arrives at a. Hence, by calculating the water consumption at cut-off, and then at release, a good idea of the amount of cylinder condensation may be obtained. If the steam used by the engine be actually caught and weighed and then compared with the weight as calculated from release an idea may be obtained of the amount of condensation at release. The computed consumption is always less than the actual consumption.

1284. Where there is a sufficient amount of compression, the work may be simplified by taking the two points a and b at the same height above the vacuum line, as shown in Fig. 278. Since the absolute pressure at a and b is the same, the clearance may be left entirely out of account, and the volume to be used in the computation will be $\frac{l}{L}$ times the volume of the cylinder, or, in other words, $\frac{l}{L} \times$ length

of stroke × area of piston. When this method is used, the steam consumption may be found directly from the following formula:

$$Q = \frac{13,750\, l\, W}{P L}, \qquad \textbf{(101.)}$$

in which Q is the number of pounds of steam consumed per

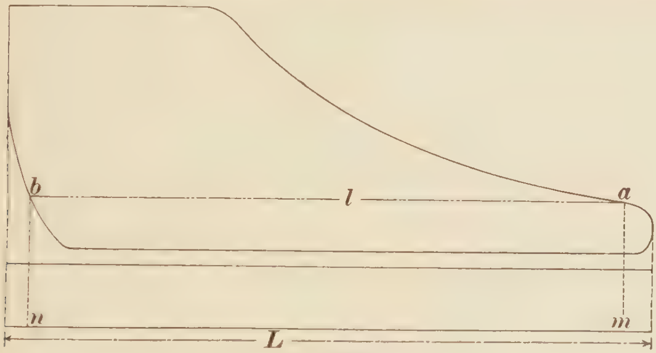


FIG. 278.

horsepower per hour, W the weight of a cubic foot of steam at the absolute pressure a, and P the M. E. P.

EXAMPLE.—From a card taken from an $18\frac{1}{2}'' \times 30''$ engine, the following measurements were obtained (see Fig. 278): $a\,m = .667$ inch; $l = 3.08$ inches; $L = 3.5$ inches; M. E. P. $= 35$ pounds. What is the steam consumption per I. H. P. per hour?

SOLUTION.—The indicator card being taken with a 45 spring, the pressure at a is $45 \times .667 = 30$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .0742 pound. Substituting in formula **101**,

$$Q = \frac{13,750\, l\, W}{P L} = \frac{13,750 \times 3.08 \times .0742}{35 \times 3.5} = 25.65 \text{ lb. Ans.}$$

EXAMPLES FOR PRACTICE.

1. Size of engine, $12'' \times 20''$; length of card L, 3.4″; length l, $2\frac{1}{4}''$; height $a\,m$, $\frac{3}{8}''$; R. P. M., 230; spring, 30; M. E. P., 18 lb. per sq. in. What is the steam consumption per I. H. P. per hour?

Ans. 25.63 lb. per I. H. P. per hour.

2. Size of engine, $12'' \times 12''$; M. E. P., 51.1; length of card L, 2.6″; length l, 1.8″; height $a\,m$, .7″; R. P. M., 350; spring, 70. What is the steam consumption per I. H. P. per hour?

Ans. 21.92 lb. per I. H. P. per hour.

3. If, in the above engine, example 2, the pressure at cut-off is 110 lb., absolute; the clearance is 8%; the length of the diagram to the point of cut-off is .7″; the pressure at a point on the compression curve is 49 lb., absolute, and the distance of this point from the end of the diagram is .14″, what is the steam consumption per I. H. P. per hour at cut-off? Ans. 19.44 lb. per I. H. P. per hour.

THE WHEELOCK ENGINE.

1285. From what has been previously stated, it should now be apparent that the clearance should be as small as may be consistent with construction, and that the valve port opening should be as large and the ports themselves as short as possible. All these conditions are fulfilled in Corliss engines. Another engine which fulfils these conditions, and has fewer parts in connection with its valve gearing, is the **Wheelock engine.** As this engine is much used in rolling mills and in other places where very heavy duty is required, and is besides a fine example of modern engine practice, a brief description of its valve gearing will now be given.

1286. Fig. 279 shows a side view of the engine complete without the fly-wheel. In the following description, *A* is the eccentric; *B* the eccentric rod; *C* the governor; *E* a lever to which the eccentric rod is attached and which alters the throw of the eccentric as in the case of Fig. 257; *D* the rod which connects this lever with the rockers *G* and *G′*, which move the valves; *H* and *H′* the crab claws; *I* and *I′* the dash-pots; *J* the pilot valve (or valve which admits and shuts off steam from the steam chest. This term is used when a second valve, called a throttle valve, is situated in the engine room, and which is used to shut off the steam from the boiler); *K* the governor rod; *M M* the floor, and *L L′* the exhaust pipes—two in this case.

In Fig. 280, a section through the valve port is given, also a top view of the valve itself. This section is taken at the back end of the cylinder, and shows the steam valve *a* shut, and the exhaust valve *b* open. The piston is, consequently, moving to the right, and the steam is flowing into the exhaust pipe *L′*. The shape of the valves is shown

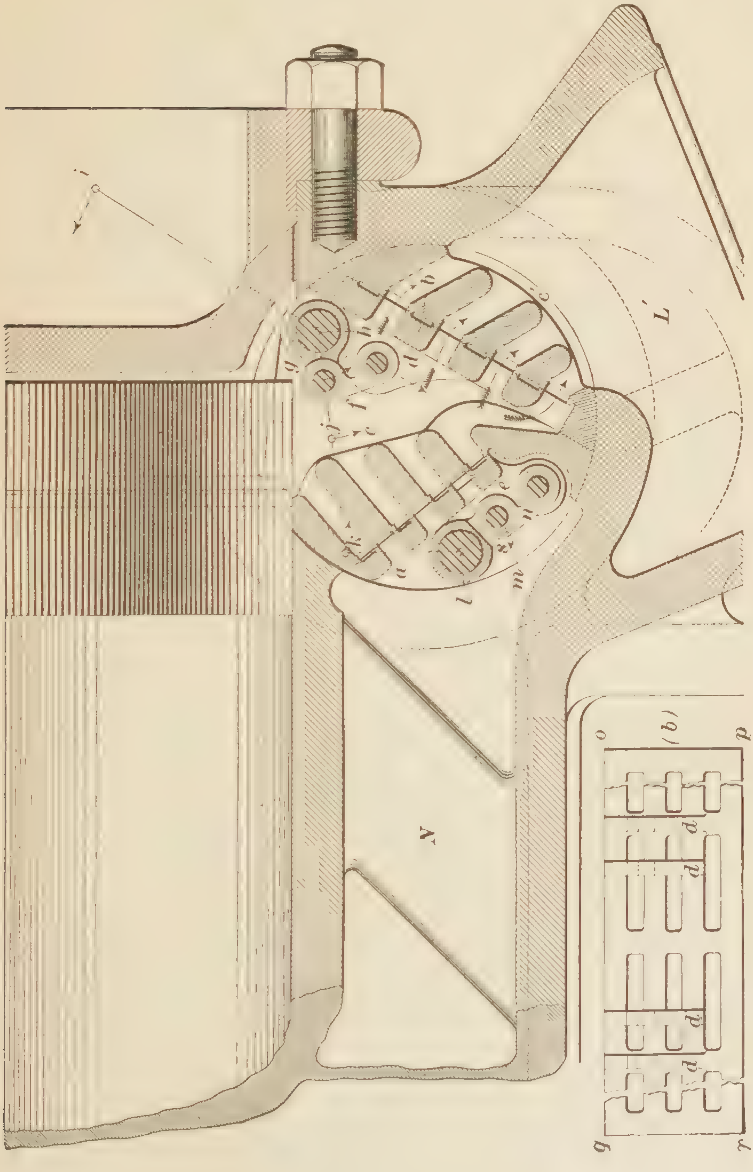


Fig. 280.

at (b). They are flat plates. The distance $o\ p$ is the *length* of the valve, and $o\ q$ the *breadth*. The breadth, as will be seen, is several times the length, and persons unfamiliar with this subject would naturally call $o\ q$ the length. The length of a valve, however, is always measured in the direction in which it moves; hence, $o\ p$ is the length. Steam is admitted to the cylinder and exhausted from it through the holes in the valves. The form of those parts of the valve marked d is clearly shown in the sectional view. Two links, f and n, are fitted between them on each valve, and secured by passing a round pin through the lugs and links. In the sectional view, c and c' are castings which form the valve seats. The ends of the valves $o\ p$ and $q\ r$ are fitted into grooves which compel any movement of the valves to be in a straight line.

Fig. 281 shows in greater detail the rocker G' and crab claw H', Fig. 279. In Fig. 280 only the center lines $i\ h$, $h\ j$, $j\ k$, and $k\ l$ have been given, in order to prevent confusion through multiplicity of lines. The pins, whose centers are h and l, are fixed in position, and the other three centers are movable. The pins h and l pass through the flange P, and to them are rigidly attached the links g and s respectively. Suppose the end i of the rocker is caused to move in the direction of the arrow; j and k will also move in the direction of the arrows. The pin h, being keyed to the rocker, will rotate in its bearing and cause the small crank g (see Fig. 280), which is keyed to h, to move downwards. This forces the link f and valve b downwards. This same movement of G' forces Q' to the right (see Fig. 281), as shown by the arrow, and thus rotates the pin l. This rotation of l causes the small crank s (Fig. 280) to move to the left, and the valve a to move upwards. Both valves continue to move in the directions shown until the point i has reached the extreme of its travel to the left, which is about 60° from its present extreme right-hand position. When G' starts on its return movement, the valves a and b also start back. Movement is imparted to G and G' by means of the eccentric A and the connections B, E, and D.

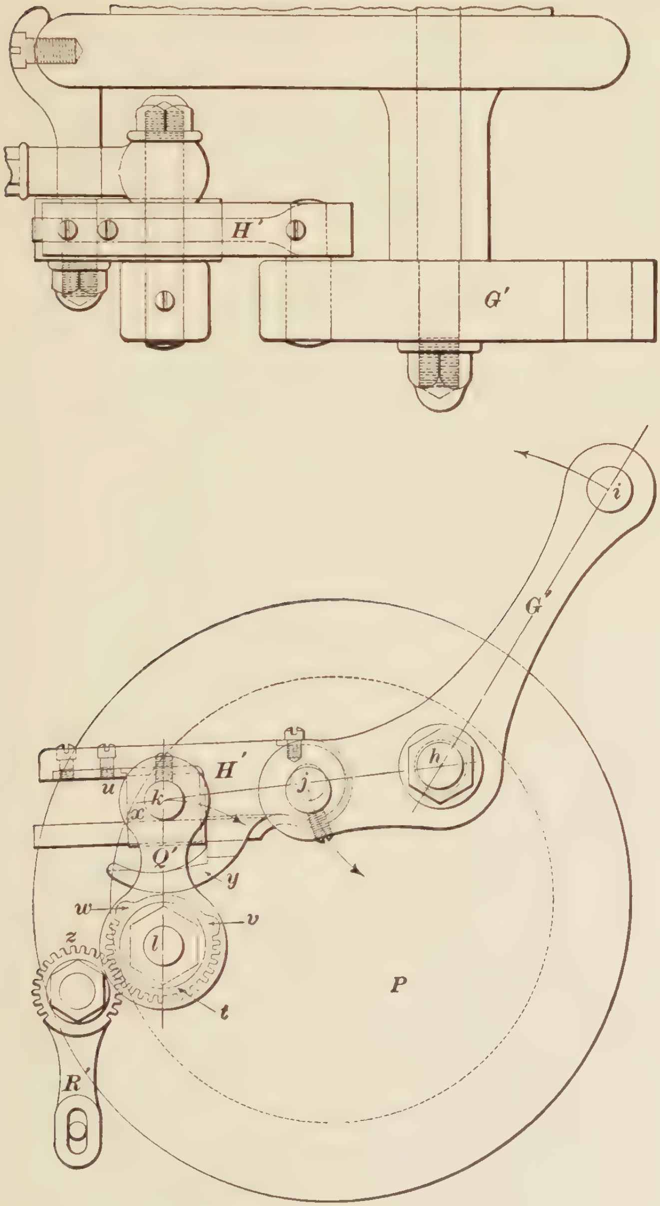


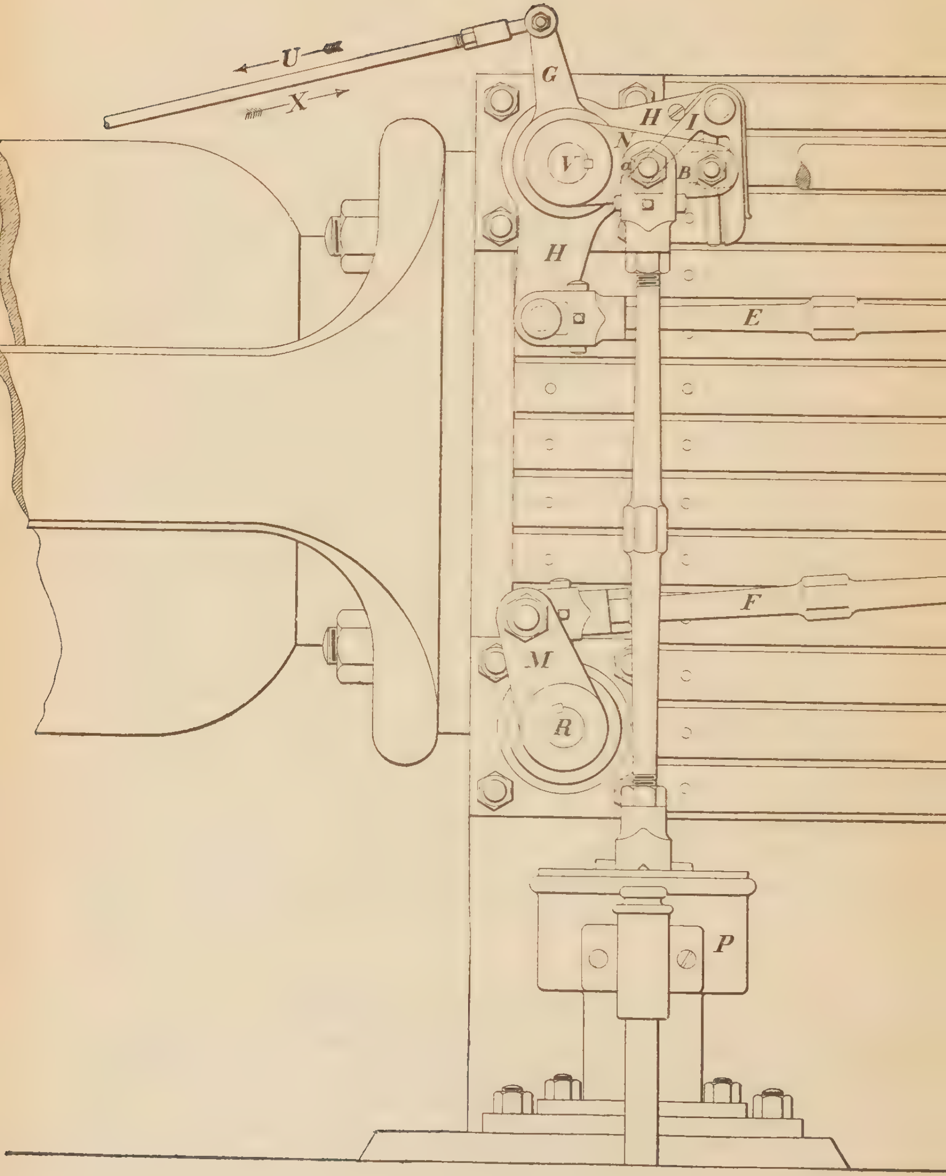
FIG. 281.

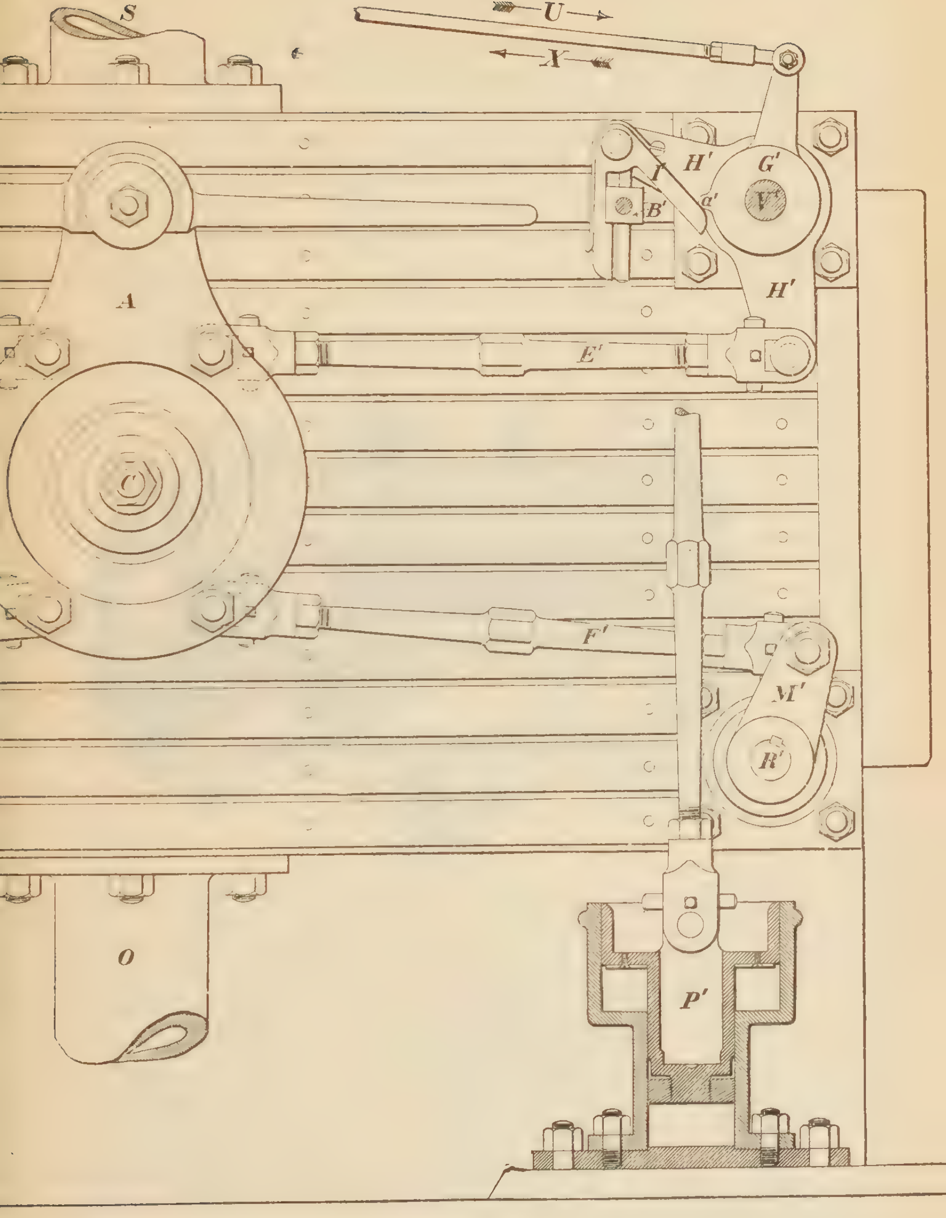
To understand the action of the governor in varying the cut-off, consider Figs. 279 and 281. In Fig. 281, the arm R' is seen to have teeth on the circumference of its upper end, which engage with another gear, t. This last gear turns loosely on the pin l, and has two projections, v and w, called *toes*. Fitting in crab claw H' is a block x, which is kept in position by the upper edge u of the crab claw. This block is connected to the dash-pot I' by means of the rod S'. Suppose that part of the load on the engine is taken off; the speed will increase, the centrifugal force of the revolving governor balls will also increase, and, in so doing, the balls fly outwards; raise a spindle, to which they are connected by means of the lower links, and cause the arm F to move to the left by means of a series of levers not shown in the figure. When F moves to the left, it carries with it the rod K, and causes R and R' to move to the left also. This movement causes the gear z to revolve to the right and the loose gear t to revolve to the left. The curved arm y of the crab claw being in a lower position than shown, owing to a movement of G' to the right, the toe v strikes this arm, raises the crab claw and disengages the block x, which is immediately pulled to the left by the dash-pot, and with it the arm Q', which actuates the valve a. The valve is then thrown to the extreme position shown in Fig. 280, and the steam is cut off from the cylinder.

When the speed of the engine is uniform, the valve cuts off in the same manner as an ordinary slide-valve, by reason of its having lap and lead. The governor varies the cut-off only when the speed of the engine changes.

CORLISS VALVE GEAR.

1287. The **Corliss valve gear** is used in a large number of engines. In Fig. 282 is shown a side elevation of this valve gear, and in Fig. 283 a section through the cylinder and valves. It has four separate and distinct valves. Two of these, v and v', Fig. 283, connect directly with the steam chest d and steam pipe s, and are called steam valves. They


U
X
G
H
I
N
V
a
B
H
E
F
M
R
P


S
U
X
H'
G'
I'
a'
V'
B'
A
H'
E'
C
F'
M'
R'
O
P'

are rigidly connected with the cranks N and N' (Fig. 282), N' being removed in order to show more clearly the disengaging link I'. The other two valves, r and r', Fig. 283, connect directly with the exhaust chest l and the exhaust pipe o, and are called exhaust valves; they are rigidly connected with the cranks M and M', Fig. 282. All the valves are cylindrical in form, and extend across the cylinder above and below, respectively.

A, Fig. 282, is a disk or wrist-plate which is made to rock upon a stud C, by the eccentric rod B, connecting it with an eccentric on the crank-shaft.

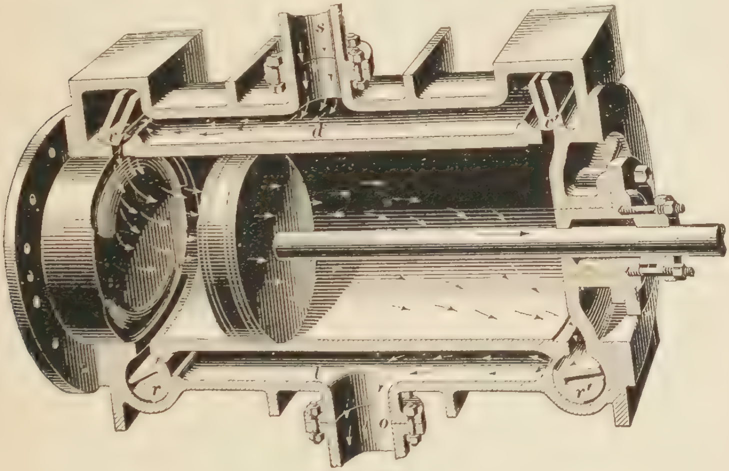


FIG. 283.

There are four valve stems: E and E', which connect the wrist-plate A with the bell cranks H and H' of the steam valves, and F and F' which connect the wrist-plate A with the cranks M and M' of the exhaust valves. The valve stems can be lengthened or shortened as the case may require, and the action of any one valve regulated independently of the other three. As the wrist-plate A rocks backwards and forwards, the exhaust valves R and R', which are rigidly connected with their cranks M and M', rock with it. The bell cranks H and H', which are provided with the disengaging links shown at I and I', are also given this rocking motion,

and by hooking on to the blocks B and B', which are rigidly connected to the cranks N and N', open the steam valves V and V'.

The projections a and a', on the two trip collars G and G', unhook these disengaging links I and I', after they have rotated the valves V and V' through a certain angle, and the cranks N and N' are pulled back to their first positions by the vacuum air dash-pots P and P', against the resistance of which the valve cranks N and N' were raised. The movements of the valves open and close the steam and exhaust ports of the cylinder at the proper intervals. The pins of the valve stems are so located on the wrist-plate that the steam valves V and V' have their quickest movement while the exhaust valves R and R' have their slowest movement, and the exhaust valves have their quickest movement while the steam valves have their slowest movement. As a consequence of this arrangement, the steam and exhaust valves have entirely independent movements, and the inlet ports may be suddenly opened full width by the quick movement of the steam valves, while the exhaust valves are practically motionless. The advantage of this valve gear is that it permits an earlier cut-off, with a greater range, a more perfect steam distribution, and a smaller clearance space than is attained with the plain slide-valve.

Engines fitted with the Corliss valve gear cannot run at much more than 90 revolutions per minute.

GOVERNORS.

1288. When a steam engine is running at a uniform speed, the work done by the steam in the cylinder must just equal the resistance overcome at the fly-wheel rim. Should the resistance become less than the work, the amount of work in excess of that necessary to overcome the resistance would cause the moving parts to move faster and faster, and the engine would "race" or "run away." If, on the contrary, the resistance should exceed the work, the engine would slow down, and finally stop. The work required of the engine

can not, of course, remain always constant; hence, it is necessary to have some means of automatically adjusting the steam supply to the variation of the resistance. This is accomplished by the *governor*.

Steam engine governors may be divided into two classes: (1) **throttling governors,** which throttle the steam in the supply pipe, and (2) **automatic** or **adjustable cut-off governors,** which regulate the steam supply by changing the point of cut-off of the valve.

1289. The ordinary throttling governor, shown in Fig. 242, consists of a balanced throttle valve placed on the steam pipe *s*; this valve is attached to the spindle *k*, at the upper end of which are the two **fly-balls** *m*, *m*, the spindle and fly-balls forming what is known as a **revolving pendulum.** The spindle and balls are driven from the main shaft by the belt *p* and the bevel wheels *o*. If the engine moves faster than the desired speed, the fly-balls are forced to revolve at a higher speed, and will, consequently, move outwards and upwards through the action of centrifugal force. This forces the spindle *k* downwards, and partly closes the throttle valve. The engine thus takes less steam, and the speed falls to the desired point, the governor balls in the meantime returning to their original position. Should the resistance become greater than the power of the engine, it slows up slightly, the balls drop and open the valve wider. More steam is admitted, and the engine immediately regains its original speed. The chief objection to the throttling governor is that the steam is wire drawn. The term **wire drawn** is applied to any case in which the steam pressure is reduced owing to the insufficiency of valve opening. The term **throttled** is also applied to such cases. Steam is more or less wire drawn in all engines fitted with plain slide valves—the rounded corners on the diagram prove this; this is because the movement of the valve is comparatively slow when closing the ports. With Corliss and other releasing gear engines, the valve movement at cut-off and release is very rapid, and the wire drawing very slight.

1290. In throttling engines (plain slide-valve engines), the steam line of the indicator diagram always inclines towards the atmospheric line instead of being nearly parallel with it, as in the cards previously given. In consequence of this, the mean effective pressure is less. Another form of a governor is shown in Fig. 284. This is the well-known Pickering governor. There are three balls, and they move outwards against the resistance of gravity and the three flat springs *s*. In so doing they lower the valves *v* and *v*′. The steam enters at *K*, flows in the direction of the arrows, and then through *J* into the steam chest. Since steam is on both sides of the valves, they are *balanced.* The object of using two valves instead of one is to afford a large opening with a small lift of the valve.

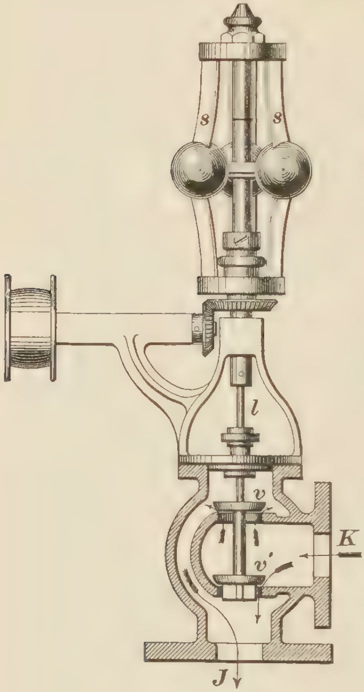


FIG. 284.

1291. As applied to the Corliss type of engines, the revolving pendulum or fly-balls vary the point of cut-off instead of throttling the steam supply. The method of operation is shown in Fig. 285; the fly-balls *m*, *m* are here given a rotary motion by a belt and gear wheels, in the same manner as in Fig. 242.

Let it be supposed that the engine is running at its proper speed. The fly-balls will then be held in their normal position by the balance existing between the centrifugal and gravity forces acting on *m* and *m*. Suppose, now, the speed of the engine increases from any cause whatever, the

centrifugal force acting on the fly-balls will also increase and will continue to pull them out, that is, to increase the diameter of the circle in which they rotate, until a new balance is effected between it and the attraction of gravity. This movement of the fly-balls will be transmitted to the lever D, causing it to turn slightly about its center in the direction of the arrow X. The movement of D will cause the trip collars G and G' (see Fig. 282) to turn through a small angle in such a direction that their projections a and a' will unhook the disengaging links I and I' earlier in the stroke. This will cause the point of cut-off to occur earlier in the stroke, and a decrease in the speed of the engine, on account of the

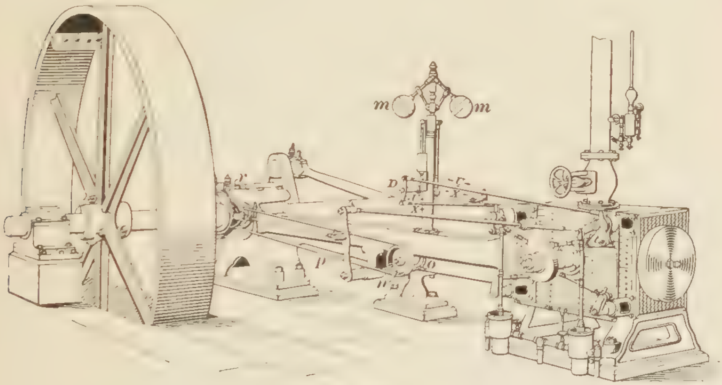


FIG. 285.

reduction in the amount of steam admitted to the cylinder, and an increased ratio of expansion of the steam under the same initial pressure. Should the speed from any cause diminish, a reverse operation would be the result. The fly-balls would drop slightly; D would turn as indicated by the arrow U, and the trip collars G and G' would be rotated in such a manner as to cause their projections a and a' to unhook the disengaging links I and I' later in the stroke; the cut-off would then occur later in the stroke, and a diminished ratio of expansion at the same pressure would again bring the speed up to its proper point.

1292. All engines which require a disengagement of some of their parts in order to affect the cut-off by the action

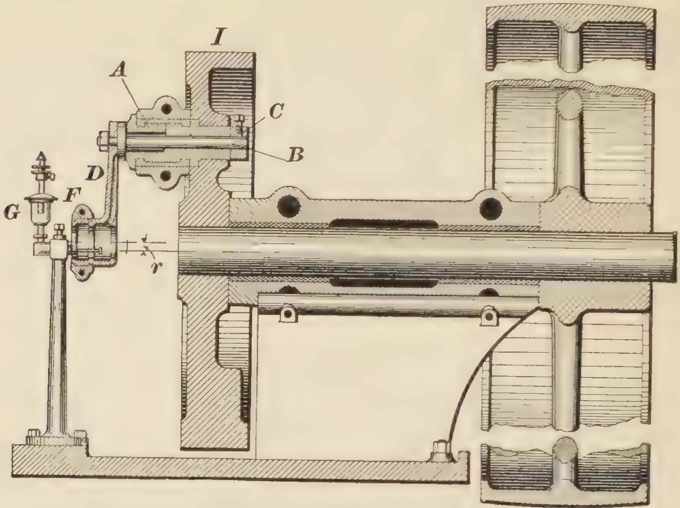


FIG. 286.

of the governor, as the Wheelock, Corliss, etc., are termed **releasing gear** engines, in contradistinction to those in which the cut-off is affected directly by the governor, without the intervention of dash-pots, cams, weights, etc. The valve gearing of this latter class is termed **positive.** Engines of the releasing gear type are limited to a speed of 90 revolutions per minute, or less, since, if the speed is increased beyond this limit, the valves will not work properly. Hence, to obtain a high piston speed, it is necessary to make the cylinders very long. Engines are sometimes required to have a high rotative speed, as in the case of electric lighting.

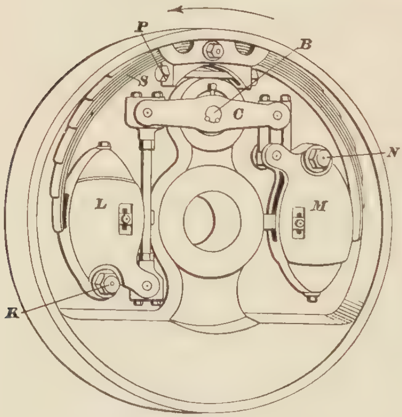


FIG. 287.

and those having positive valve gears respond fully to all demands. By **high rotative speed** is meant a large number of revolutions per minute. The rotative speed of a positive geared engine may be 400 revolutions per minute, or more, and still be sensible to variations in the load.

In automatic cut-off engines having positive valve gears, the governors are of the type known as **shaft governors.** There are many varieties of the shaft governor, but only one will be described here.

Fig. 286 shows a section through the shaft and fly-wheels, and Fig. 287 a perspective view of the fly-wheel to which the governor is attached. Engines of this class usually have two fly-wheels—one termed the *governor wheel* and the other the *belt* or *band wheel.* If necessary, both wheels can drive belts, but it is not usual to use the governor wheel for this purpose, unless it is the only fly-wheel. By reference to Fig. 286, it will be seen that the crank pin A is hollow, and that a rod B passes through it, to which is attached a crank D, on the left end. This crank has an enlarged end and really forms the eccentric, F being the eccentric sheave. The manner in which the enlarged end of D forms the eccentric is this: The center line of the enlarged end, and, consequently, of the sheave F, does not coincide with the axis of the engine shaft; hence, since the arm D is always in the relative position in regard to a point on the governor wheel, unless the governor acts in a manner to be hereafter described, the distance r between these two center lines remains the same as the crank and governor wheel revolve. The distance r is the eccentric radius, and $2r$ is the throw of the eccentric = the travel of the valve.

In Fig. 287, C is a bar keyed to the rod B which passes through the crank-pin. On each end of C is suspended a rod; both of the rods are attached to the heavy weights L and M, which are pivoted at K and N. S is a finely tempered spring which presses against the weights. These two weights and rods, the bar C and spring S, form the governor, which acts in the following manner: As the governor wheel

revolves in the direction of the arrow, the centrifugal force generated in the weights *L* and *M* tends to cause them to turn on their pins *K* and *N*, but this motion is resisted to a certain extent by the spring *S*. The spring holds the weights in equilibrium against the centrifugal force when the speed of the engine has reached the desired limit. For a speed beyond this limit, the centrifugal force overcomes the force of the spring and the weights move outwards; but, for a speed below the limit, the force of the spring overcomes the centrifugal force and the weights move inwards. The eccentric is never in the position shown in Fig. 286 except when the engine is not running; it usually occupies a position such that the distance *r* is greater when the engine is running. Suppose the engine is running at its rated speed under its rated load. The weights and the eccentric will occupy a certain position, and the valve will travel back and forth an amount corresponding to twice the distance *r*, whatever that may be. All the operations will be exactly the same as in the case of the ordinary slide valve with a fixed eccentric. Suppose that an extra load is thrown upon the engine. The speed will momentarily slacken; the weights will move inwards; the left end of the bar *C* will turn towards the shaft and the right end away from the shaft. This causes the rod *B* to rotate in its bearing in such a manner that the arm which carries the eccentric is forced further away from the shaft; the distance *r* is increased and, consequently, the valve travel; the cut-off is later; more steam is admitted, and the engine almost instantly regains its former speed, the weights returning to their former position. Should the speed be increased, the reverse takes place. The balls *L* and *M* fly outwards and rotate the lever *C*, rod *B*, and arm *D* in the opposite direction; the distance *r* is lessened; the cut-off takes place earlier, and the engine regains its former speed.

The speed of the engine may be varied by means of the nut *P*. By tightening or loosening *P*, the tension of the spring is altered, and with it the speed of the engine.

G, Fig. 286, is an oil cup which is supported by the stand *E*. The oil drops into the hollow eccentric and flows down the arm *D*, when it is revolving, into the bearing, lubricating the rod *B* so that the governor may work freely.

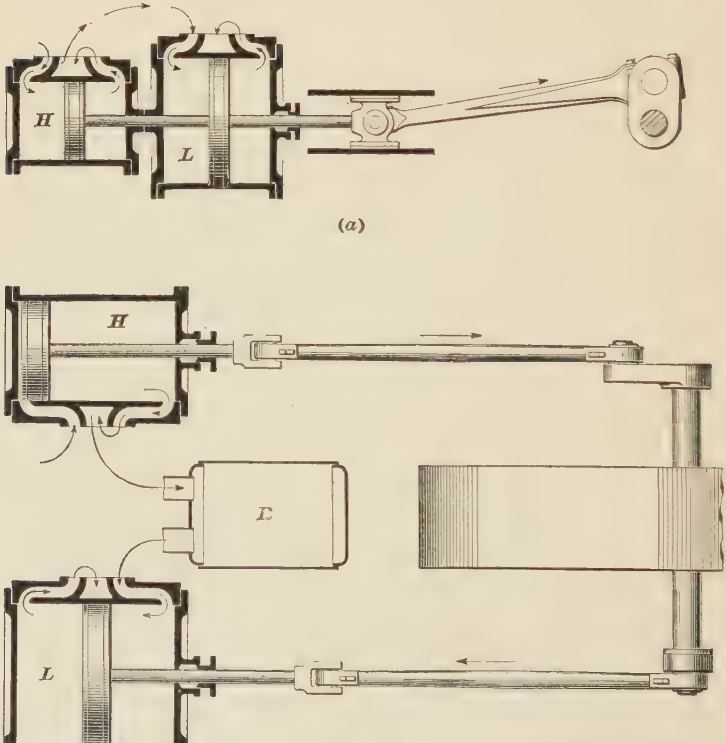
COMPOUND ENGINES.

1293. The thermal efficiency of the steam engine has been shown to be $\frac{T_1 - T_2}{T_1}$. T_2 can not, in practice, be lower than the absolute temperature of the condenser. Hence, the only way of increasing the fraction $\frac{T_1 - T_2}{T_1}$ is to increase T_1, or, in other words, to increase the temperature and, consequently, the pressure of the entering steam. Following out this idea, steam pressures have steadily increased from 8 to 10 pounds, in the time of Watt, to 150 to 200 pounds per square inch, the pressures used in modern locomotives and marine engines. But here the evil of cylinder condensation again appears; for, by increasing the range of temperature, $T_1 - T_2$, the loss by cylinder condensation is largely increased. To see this clearly, let the pressure of the steam passing into the condenser be 4 pounds above vacuum; its temperature is about 153°. Let the pressure of the entering steam be say 60 pounds above vacuum; its temperature is about 293°. The fall in temperature is 293° − 153° = 140°, nearly. Suppose, however, the entering steam has a pressure of 200 pounds above vacuum; its temperature would then be 382°, nearly, and the fall in temperature during the stroke would be 382° − 153° = 229°. Now, it is plain that a great deal more of the incoming steam must condense to raise the temperature of the cylinder walls back from 153° to 382°, than to raise them from 153° to 293°. *Hence, increasing the range of temperature increases the loss due to cylinder condensation.*

1294. To obtain the advantages of a high pressure, and, at the same time, avoid the loss due to cylinder condensation as much as possible, the steam may be allowed to ex-

pand successively in two or more cylinders. The fall of temperature is thus divided between the two or more cylinders and, consequently, the loss from condensation in both, or all of them, is made considerably less than it would be if the same fall of temperature was allowed to take place in one cylinder. When the expansion takes place in two cylinders, the engine is said to be **compound**; if the expansion takes place in three cylinders, the engine is said to be **triple expansion,** and if in four cylinders, **quadruple expansion.**

1295. Compound engines are usually made in one or the other of the two types shown in Fig. 288. In (*a*) the



(*b*)
FIG. 288.

two cylinders are placed in line, the two pistons being attached to the same piston rod. *H* is the cylinder which first

receives steam from the boiler; it is called the **high-pressure** cylinder. After the steam has expanded in *H*, it passes to the larger cylinder *L*, which is called the **low-pressure** cylinder; from here the steam is exhausted into the atmosphere or into a condenser.

Fig. 288 (*b*) shows what is known as the **receiver** compound engine. The steam enters the high-pressure cylinder *H* from the boiler; exhausts into a separate vessel *R*, called the **receiver**; from there it passes to the low-pressure cylinder *L*, and finally exhausts into the atmosphere or into a condenser.

A receiver compound engine has two piston rods and two cranks; the cranks may be placed at any angle with each other. The compound engine, without a receiver, may have one piston rod and crank, as shown in the tandem type, or it may have two piston rods and two cranks, the cylinders being placed side by side. In any compound engine, without a receiver, the two pistons must begin and end their stroke at the same time, and the two cranks must be together, or placed 180° apart.

1296. When one cylinder is placed behind the other, as shown in Fig. 288 (*a*), the engine is called a **tandem compound.** When the cylinders are placed side by side, as shown in (*b*), and the piston rods are attached to separate cross-heads, the engine is called a **cross compound**; if both piston rods are attached to the same cross-head, the engine is called a **twin compound.** If any of these types of engines have a condenser, they are called **tandem, cross,** or **twin, compound condensing engines.** Without a condenser, they are called **non-condensing engines.** They all may or may not have a receiver.

1297. In giving the size of a multiple-expansion engine, the stroke is always written last. Thus, a compound engine whose high-pressure cylinder was 11″ in diameter, low-pressure cylinder 20″ in diameter, and stroke 15″ would be expressed as a 11″ and 20″ × 15″ compound. In the same manner a 14″, 22″, and 34″ × 18″ triple-expansion engine

would indicate that the diameters of the cylinders were 14″, 22″, and 34″, and that they had a common stroke of 18″.

1298. The **ratio of expansion** of a compound or triple-expansion engine is the ratio between the volume of steam exhausted into the atmosphere, or into the condenser, per stroke, and the volume of steam in the high-pressure cylinder at the point of cut-off.

Let e = ratio of expansion in high-pressure cylinder;
" E = total ratio of expansion;
" v = volume of cylinder receiving steam from the boiler;
" V = volume of cylinder exhausting into atmosphere, or condenser.

Then, $E = \frac{eV}{v}$; **(102.)**

that is, the **total ratio of expansion,** or, as it is usually expressed, the **number of expansions,** is equal to the ratio of expansion of the small cylinder multiplied by the ratio between the volumes of the two cylinders. The total ratio of expansion in a compound engine depends only upon the relative volumes of the cylinders, and the point of cut-off in the high-pressure cylinder; it does *not* depend at all upon the point of cut-off in the low-pressure cylinder. The number of expansions in a compound engine varies from 6 to 12; in a triple-expansion, from 10 to 25.

EXAMPLE.—It is desired to have a total ratio of expansion of 9; the number of expansions in the high-pressure cylinder is 2.72; the volume of the high-pressure cylinder is 6 cu. ft. What must be the volume of the low-pressure cylinder?

SOLUTION.—$E = 9$; $e = 2.72$; $v = 6$.

Substituting in formula **102,** $9 = \frac{2.72\,V}{6}$, or $V = \frac{6 \times 9}{2.72} = 20$ cu. ft., nearly. Ans.

EXAMPLE.—The low-pressure cylinder is four times as large as the high-pressure cylinder, and the real cut-off of the latter is $\frac{2}{5}$. What is the total ratio of expansion?

SOLUTION.—
$$e = \frac{1}{k} = \frac{1}{\frac{2}{5}} = 2.5.$$
$$E = \frac{eV}{v} = \frac{2.5 \times 4}{1} = 10. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A compound engine has cylinders 15″ and 25″ diameter by 20″ stroke; that is, its size is 15″ and 25″ × 20″. The clearance in the high-pressure cylinder is 14%, and the apparent cut-off is $\frac{1}{4}$. What is the number of expansions? Ans. 6.69.

2. A 28″, 48″, and 74″ × 60″ triple-expansion engine cuts off at $\frac{3}{5}$ stroke in the high-pressure cylinder. Clearance in the high-pressure cylinder, 2%. Find the number of expansions. Ans. 17, nearly.

3. If the 26″, 40″, 60″, and 70″ × 72″ quadruple-expansion engine at the World's Fair cut off the steam in the high-pressure cylinder at $\frac{1}{4}$ stroke, and the clearance in that cylinder was 3%, what was the total number of expansions? Ans. 26$\frac{2}{3}$, nearly.

COMPOUND ENGINE DIAGRAMS.

1299. In Fig. 289 are shown the ideal diagrams of a tandem compound engine, neglecting clearance and com-

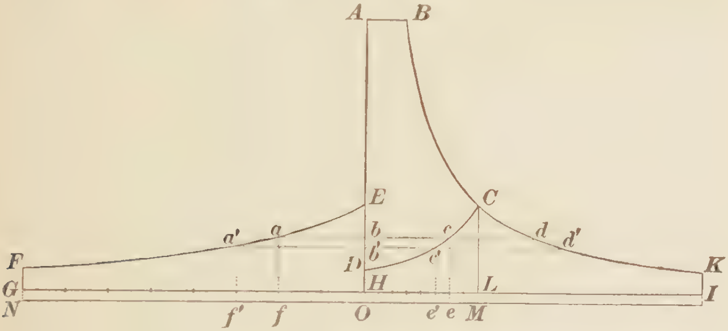


FIG. 289.

pression. $A\ B\ C\ D$ is the diagram from the high-pressure cylinder. Steam enters at the boiler pressure $O\ A$; at B, cut-off occurs, and the steam expands along the line $B\ C$ to the end of the stroke. The steam and expansion lines $A\ B$ and $B\ C$ are precisely like those of a simple engine. At C, the exhaust opens. Instead, however, of exhausting into the atmosphere or a condenser, the steam exhausts into the low-pressure cylinder. The low-pressure cylinder is always larger than the high-pressure cylinder; consequently, the volume of steam exhausting from the high-pressure into the low-pressure cylinder is constantly increasing. This is shown in Fig. 290, when the two pistons are at the end of the stroke, as shown at (a), the

steam simply fills the small cylinder *H*; but at the middle of the stroke, as shown at (*b*), the steam fills half of (*a*) and also half of (*b*). Hence, its volume grows greater as the two pistons move to the right.

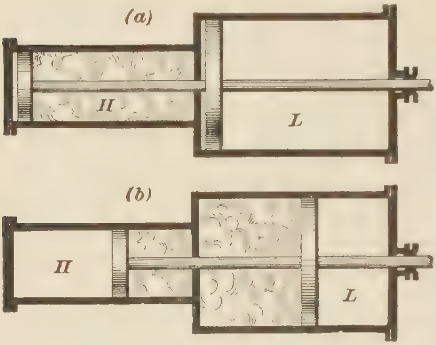


FIG. 290.

1300. Returning now to Fig. 289, the steam, when the piston is at the end of the stroke *c*, just fills the small cylinder; on the return stroke, however, as has just been shown, the volume increases, and hence the pressure falls, as shown by the back pressure line *C D*. At the end *D* of the return stroke, steam again enters the right side of the cylinder *H* from the boiler and raises the pressure to *A*, thus completing the cycle of operation.

C D H L represents the card from the low-pressure cylinder *L*. Since the high-pressure cylinder exhausts directly into the low-pressure cylinder, the back pressure of the former must be the same as the forward pressure of the latter. Hence, *C D* is both the back pressure line of *H* and the expansion line of *L*. At the end of the stroke, the pressure drops from *D* to *H*, *O H* being the pressure in the condenser. The remainder of the diagram is the same as that of a simple engine. In the above description, the common length *O M* of the two diagrams has been taken proportional to the length of stroke.

Suppose now that the length of each diagram be taken to represent the volume of the cylinder to which it belongs. Let *O M* represent the volume of the high-pressure cylinder. Suppose that the volume of the low-pressure cylinder is say 3 times the volume of the high-pressure cylinder. The length of the low-pressure diagram must then be 3 times that of the high-pressure diagram. From *O* lay off *O N* =

$3 \times O\ M$, and project the expansion line $C\ D$ into the line $E\ F$. This is done by dividing the line $O\ M$ into a convenient number of equal parts, in this case 8, and $O\ N$ into the same number of equal parts. At the points of division e, e', etc., erect ordinates cutting $C\ D$ in c, c', etc. At the points of division O, N erect ordinates $f\ a$, $f'\ a'$, etc. Through c, c', etc., draw lines parallel to $M\ N$, cutting $a\ f$ in a, $a'\ f'$ in a', etc.; a, a', etc., will be points on the required line $E\ F$. E and F are, of course, opposite C and D. $E\ F\ G\ H$ represents the low-pressure card to the same scale of pressures and volumes that $A\ B\ C\ D$ represents the high-pressure card—$E\ F\ G\ H$ has been laid off to the left of $O\ A$ simply for convenience.

1301. The two diagrams may be combined into one in the following manner: Draw a horizontal line, as $a\ d$, intersecting both diagrams. The volume of steam in the high-pressure cylinder at the pressure $e\ c = O\ b$ is represented by the length $b\ c$; the volume of steam in the low-pressure cylinder at the same pressure is represented by $a\ b$. Hence, the total volume of steam at the pressure in question is $a\ b + b\ c = a\ c$. From c, lay off $c\ d = a\ b$; then, $a\ b + b\ c = b\ c + c\ d = b\ d =$ volume of steam at pressure $O\ b$. In the same manner find that $b'\ d'$ equals the volume of steam in both cylinders when the pressure is represented by $O\ b'$. By finding a sufficient number of these points, d, d', etc., the curve $C\ K$ may be drawn. This curve represents the relation between the common pressure in the two cylinders and the total volume of steam in both cylinders, and it will be found that it is simply a continuation of the expansion curve $B\ C$ of the high-pressure cylinder. It is seen that the combination of these two diagrams forms one large diagram, $A\ B\ C\ K\ I\ H\ A$, equal in area to the sum of the areas $A\ B\ C\ D$ and $E\ F\ G\ H$; for the mean ordinates of $E\ F\ G\ H$, $C\ D\ H\ L$, and $C\ K\ I\ L$ are equal from the nature of the construction; hence, representing the mean ordinate by h, the area of $E\ F\ G\ H = h \times G\ H$; of $C\ D\ H\ L = h \times H\ L$, and of $C\ K\ I\ L = h \times L\ I$. But $G\ H = 3\ H\ L$ and $L\ I = 2$

$H\ L$; consequently, area $E\ F\ G\ H = h \times 3\ H\ L = h \times H\ L + h \times 2\ H\ L =$ area $C\ H\ I\ K$. The length of the large diagram represents the volume of the low-pressure cylinder, while the initial pressure $O\ A$ is the boiler pressure of the steam entering the high-pressure cylinder.

The ratio of expansion of the compound engine of Fig. 289 is $\frac{H\ I}{A\ B}$. If the large diagram $A\ B\ K\ I\ H\ A$ be considered as the diagram of a single engine, the ratio of expansion is also $\frac{H\ I}{A\ B}$. A single engine giving the large combined diagram $A\ B\ K\ I\ H\ A$ will do the same work as the compound engine giving the two diagrams $A\ B\ C\ D$ and $E\ F\ G\ H$; but, in order that a single engine may give the diagram $A\ B\ K\ I\ H\ A$, the volume of its cylinder must be represented by $H\ I$—that is, it must be equal to the volume of the low-pressure cylinder of the compound engine, and, further, it must work with an initial pressure $O\ A$ equal to the boiler pressure of the compound.

The horsepower of a compound engine is approximately equal to the horsepower of a single engine having a cylinder equal in volume to low-pressure cylinder of the compound, and working with the same ratio of expansion and with the same boiler pressure.

In general, it is customary to calculate the horsepower of a compound, triple, or quadruple-expansion engine as if the total expansion took place in the cylinder exhausting into the condenser. This will give a rough approximation to the true horsepower, which will usually be less than the calculated value.

EXAMPLE.—The low pressure cylinder of a compound engine is 30″ × 40″. The boiler pressure is 100 pounds (gauge) and the number of expansions 8. Find the approximate horsepower, assuming the number of revolutions per minute to be 60.

SOLUTION.—Absolute pressure $P = 100 + 14.7 = 114.7$ lb.; take back pressure p, equal to 3 lb. for condensing engine; ratio of expansion $r = 8$.

Substituting in formula **99**,

$$\text{M. E. P.} = \frac{.9\,P\,(1 + 2.3 \log r)}{r} - .9\,p =$$

$$\frac{.9 \times 114.7\,(1 + 2.3 \log 8)}{8} - 9 \times 3 = 37 \text{ pounds, nearly.}$$

Now, using formula **98**,

$$\text{I.H.P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{37 \times 40 \times 30^2 \times .7854 \times 60 \times 2}{33{,}000 \times 12.} = 317.016 \text{ H.P. Ans.}$$

1302. Fig. 291 shows the ideal diagram of a tandem compound engine, taking clearance and compression into

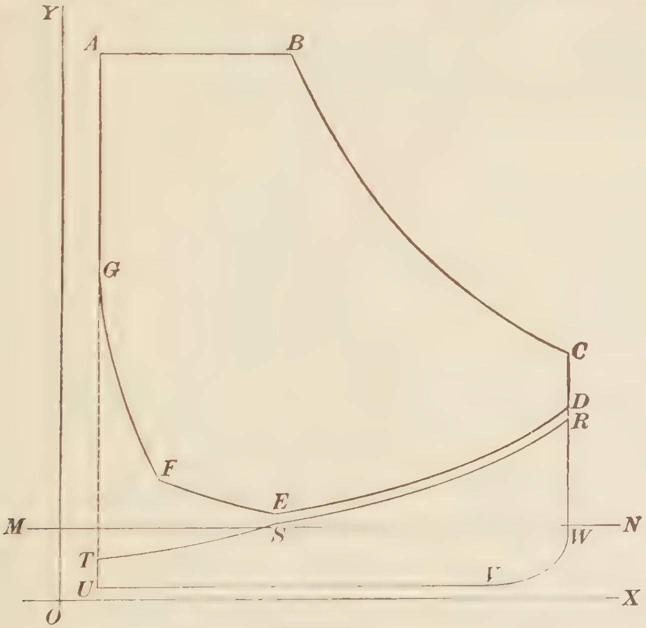


FIG. 291.

account. The steam and expansion lines $A\ B$ and $B\ C$ of the high-pressure cylinder are similar to those of a single engine. At C the pressure drops slightly as the steam is admitted to the low-pressure cylinder. The back-pressure line of the high-pressure card and the expansion line of the low-pressure card are parallel, the slight difference between them being due to the resistance of the pipe connecting the cylinders. At E the steam is cut off from the large cylinder,

and is compressed in the small cylinder, and in the pipe connecting the two. At F the exhaust closes, and the steam is compressed in the small cylinder alone from F to G; at G fresh steam again enters.

When the cut-off of the low-pressure cylinder occurs as at S, the steam already in the cylinder expands, following the ordinary equilateral hyperbola $S\ T$. At T release takes place, and the pressure falls to the pressure of the condenser. The remainder of the card is the same as for a simple engine.

1303. In Fig. 292 is shown a diagram taken from an actual engine. The similarity between it and the theoretical diagram of Fig. 291 will readily be seen.

FIG. 292.

1304. The action of the steam in a receiver cross-compound engine may perhaps be better shown by assuming certain conditions to be fulfilled, and from them working out the theoretical diagram.

Suppose that the volume of the low-pressure cylinder is 12 cu. ft.; that the volume of the high-pressure cylinder is 4 cu. ft., and that the volume of the receiver is also 4 cu. ft.; that the steam is to be cut off in the high-pressure cylinder

at $\frac{1}{3}$ stroke, and in the low pressure cylinder at $\frac{1}{2}$ stroke; that the boiler pressure is 100 pounds absolute, and the cranks make an angle of 90° with each other. Neglect clearance and compression. The stroke of the high-pressure cylinder begins at A (see Fig. 293), the pressure at A being 100 lb.; at B cut-off takes place, and the steam expands along the equilateral hyperbola $B\,C$. The volume of steam at C is 3 times that of B, and, since $p_1 v_1 = p_2 v_2$, the pressure at C is $\frac{100 \times 1}{3} = 33.3$ pounds. The steam is released at C, and passes into the receiver, where it mixes with steam at receiver pressure. In order to find the resulting pressure of the mixture of the steam in the high-pressure cylinder, and

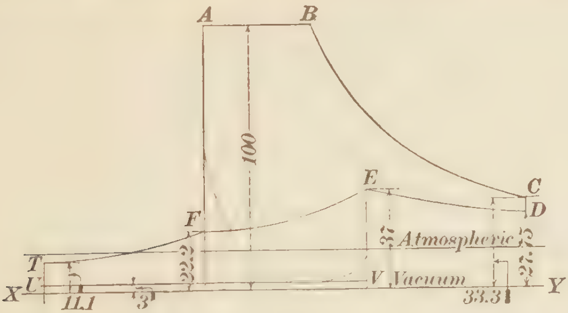


FIG. 293.

the steam in the receiver, it will first be necessary to find the pressure of the latter. The total ratio of expansion is found as follows: The volume of the high-pressure cylinder at cut-off is $\frac{4}{3}$ cu. ft.; volume of low-pressure cylinder is 12 cu. ft.; number of expansions $= 12 \div \frac{4}{3} = 9$. If clearance had been taken into account, the above would be modified to a certain extent. It can be proven that the receiver exerts no effect whatever upon the number of expansions or upon the final terminal pressure. This is true, no matter what the size of the receiver may be.

Hence, the terminal pressure in the low-pressure cylinder is $\frac{100}{9} = 11.1$ pounds. Since the low-pressure cylinder cuts off at $\frac{1}{2}$ stroke, the pressure at the point of cut-off is

$11.1 \times 1 = p \times \frac{1}{2}$, or $p = 22.2$, the volume of steam at cut-off being half that at end of stroke. Now, just before cutting off, the low-pressure cylinder was receiving steam from the receiver; hence, the pressure in the receiver at the instant of cut-off in the low-pressure cylinder is 22.2 pounds. Since the cranks are at right angles to each other, the high-pressure piston is just at the end of its stroke when cut-off occurs in the low-pressure cylinder. Hence, the steam at 33.3 pounds pressure, on being released from the high-pressure cylinder, rushes into the receiver and mixes with the steam at 22.2 pounds pressure. The pressure of the mixture is found from formula **63,** Art. **1062,**

$$V P = v p + v_1 p_1.$$

The volume of steam in the high-pressure cylinder having a pressure of 33.3 lb. is 4 cu. ft., and that in the receiver having a pressure of 22.2 lb. is 4 cu. ft. The low-pressure cylinder being cut off, the total volume of the mixture is 8 cu. ft.

Substituting, $8 \times P = 4 \times 22.2 + 4 \times 33.3$, or $P = 27.75$ lb. The pressure thus drops from C to D. The steam is now compressed in the high-pressure cylinder and receiver by the return stroke of the high-pressure piston. The volume of steam at D is 8 cu. ft.; when the piston has completed one-half of its return stroke the volume is $4 + 2 = 6$ cu. ft.; hence, the pressure is $27.75 \times \frac{8}{6} = 37$ lb., as shown at E. Since at E the high-pressure piston is at the middle of its return stroke, the low-pressure piston must be at the beginning of its return stroke. The steam in the receiver and high-pressure cylinder is now admitted to the other end of the low-pressure cylinder, its volume increases as shown in Fig. 290, and its pressure falls accordingly, as shown by the line $E F$, which is a part of the back-pressure line of the high-pressure diagram, and the steam line of the low-pressure diagram. The pressure at F has already been found to be 22.2 pounds, and the terminal pressure at T to be 11.1 pounds. Hence, the expansion line of the low-pressure diagram is an equilateral hyperbola through F and T. At T

the pressure drops to that of the condenser, about 3 pounds; the remainder of the card is the same as for a simple engine.

In Fig. 294 is shown a diagram from a compound marine

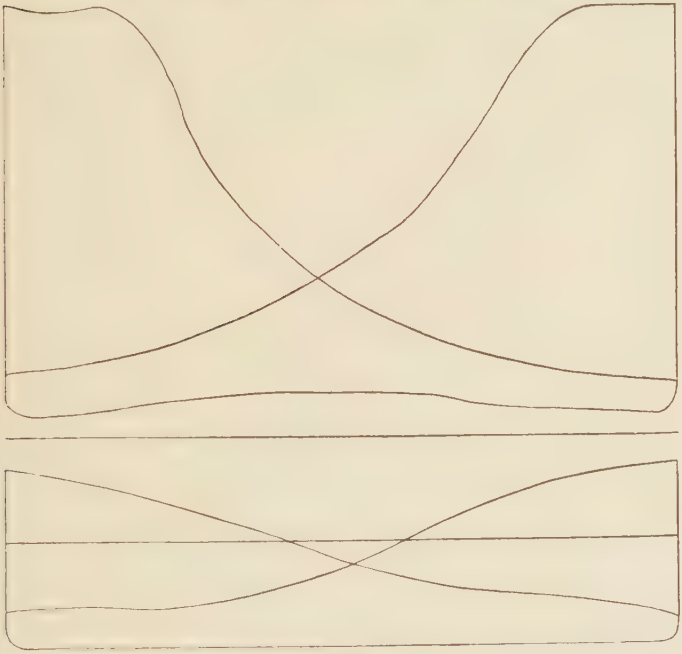


FIG. 294.

engine. It will be readily seen that it quite closely resembles the theoretical diagram.

USUAL METHOD OF COMBINING THE CARDS.

In order to show the usual method of combining the cards of a multiple (two or more) expansion engine, Figs. 294, 295, and 296 have been drawn. In Fig. 295, *H* is a diagram from the high-pressure cylinder of a Porter-Allen tandem-compound condensing engine, and *L* is a diagram from the low-pressure cylinder. When taking cards from compound or other multiple-cylinder engines, it is usual to use different springs in each indicator on the different cylinders in order to have the height of the cards as nearly alike as possible.

since the larger the diagram, the more clearly can the various points about the diagram be seen. The lengths of all the cards are made the same, if possible. In the present case the lengths of both cards are nearly the same, but a 60 spring was used to take the card *H*, and a 20 spring for the card *L*. Reducing *L* to a 60 spring by dividing the atmospheric line *A B* into 10 equal parts, and making the ordi-

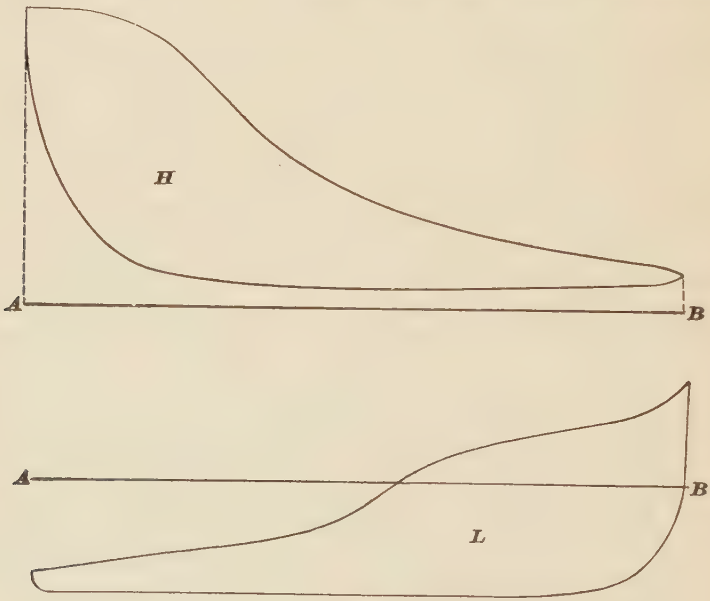


FIG. 295.

nates through these parts $\frac{20}{60} = \frac{1}{3}$ as long, Fig. 296 is obtained by simply placing the atmospheric line of one card upon that of the other.

1305. In order to study the effects of the expansion, and to determine whether the points of cut-off have been properly located, the two diagrams shown in Fig. 296 must be combined into one diagram. It should be noted that the two diagrams in Fig. 296 are precisely the same as if both had been taken with a 60 spring, and one laid upon the other so that their atmospheric lines coincided. The necessary

engine data are as follows: Diameter of high-pressure cylinder, 24 in.; of low-pressure cylinder, 46 in.; stroke, 42 in.; clearance of high-pressure cylinder, 7%; of low-pressure cylinder, 2%. From this it is seen that the volume of the

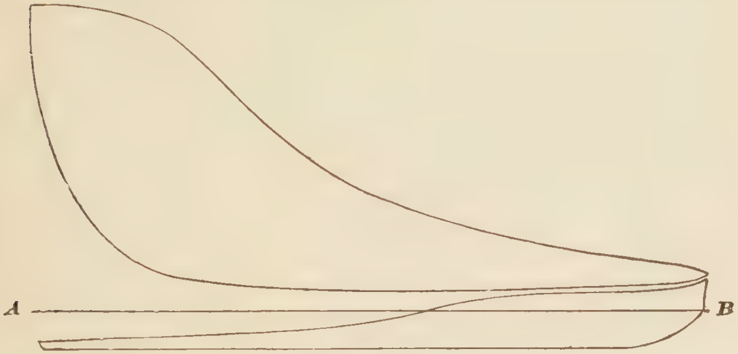


FIG. 296.

low-pressure cylinder (piston displacement) is 3.66, say $3\frac{2}{3}$ or $\frac{11}{3}$ times that of the high-pressure cylinder. The actual volume of the clearance in both cylinders is about the same, since $.07 \div \frac{11}{3} = .02\frac{1}{3} = 2\%$ of the low-pressure cylinder vol-

FIG. 297.

ume, nearly. In order to prevent the great length of the cards, by increasing the length of the low-pressure card an amount corresponding to the ratio of the volume of the low-pressure to the high-pressure cylinders, it is usual to have

the low-pressure card as shown in Fig. 296, and *decrease* the length of the high-pressure card an amount corresponding to the ratio of the cylinder volumes. This has been done in Fig. 297, and in the following manner: Divide A B in H, Fig. 295, into 10 equal parts and erect ordinates at the points of division. Make C B, Fig. 297, equal in length to $\frac{3}{11}$ of A B, same figure, and divide C B into 10 equal parts. Draw ordinates through the points of division on C B, and make them equal in length to the corresponding ordinates on H,

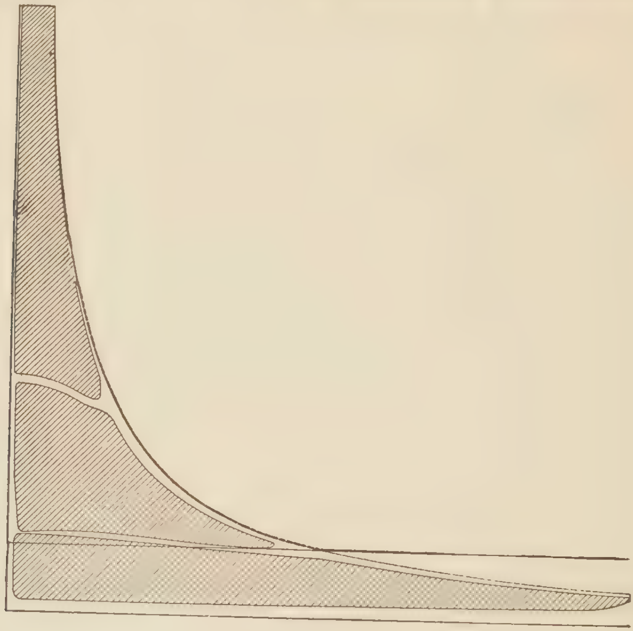


FIG. 298.

Fig. 295. Trace a line through the extremities of these ordinates, and the result will be D E F G D, or the high-pressure diagram reduced to the volume of the low-pressure cylinder. E is the point of cut-off in the high-pressure cylinder, I K the clearance line (D K being 2% of A B), and H I is the vacuum line. With I as the point of no pressure and no volume, pass an isothermal curve through the point of cut-off E, thus obtaining the dotted line J E L. If there

were no losses, the area $DJLNMD$ would represent the work done by the steam, and the difference between this area and the area of the shaded portions represent, approximately, the various losses. As will be seen, this loss is comparatively small, and could be made still smaller by making the cut-off in the low-pressure cylinder a trifle later.

In a manner entirely similar to that just described, the cards of a triple or quadruple expansion engine are combined. Fig. 298 shows the combined cards taken from a Reynolds-Corliss triple-expansion condensing pumping engine. The diameters of the cylinders were 28, 48, and 74 in., and the stroke was 60 in.; the clearance was 1.4%, 1.5%, and .77%, respectively.

1306. Ratio of Cylinders.—The ratio between the volumes of the two cylinders is so chosen that: 1, the power is divided as equally as possible between them; 2, the initial strains in the two cylinders may be the same; 3, the drop of pressure between the high-pressure cylinder and the receiver may be as small as possible. Numerous rules and formulas are used for this purpose. One rule is to make the number of expansions in the high-pressure cylinder 2.72. Substituting this in formula **102**,

$$E = 2.72\frac{V}{v}, \text{ or } \frac{V}{v} = \frac{E}{2.72}. \qquad \textbf{(103.)}$$

EXAMPLE.—What should be the ratio between the volumes of the two cylinders if the total number of expansions be 10?

SOLUTION.— $\frac{V}{v} = \frac{E}{2.72} = \frac{10}{2.72} = 3.68.$ Ans.

Another rule is to make the required ratio equal to the square root of total ratio of expansion; that is,

$$\frac{V}{v} = \sqrt{E}. \qquad \textbf{(104.)}$$

Using the formula in the last example,

$$\frac{V}{v} = \sqrt{E} = \sqrt{10} = 3.16. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A compound engine is to be designed. The high-pressure cylinder is to be 17″ × 20″, with an apparent cut-off of .4, and a clearance of 10%. The total number of expansions is to be 7. What must be the size of the low-pressure cylinder having the same stroke?

Ans. 30.324″ × 20″, say $30\frac{1}{4}$″ × 20″.

2. Calculate the above by formula **103.** Ans. $27\frac{1}{4}$″ × 20″.

3. The low-pressure cylinder of a compound engine is 44″ × 36″. What must be the diameter of the high-pressure cylinder in order that there may be 8 expansions, with .38 cut-off and 12% clearance in the high-pressure cylinder? Ans. 23.28″, say $23\frac{1}{4}$″.

4. An 11″ and $22\frac{1}{2}$″ × 18″ compound engine has 13% clearance in the high-pressure cylinder. Find the point of apparent cut-off so that there may be 9 expansions. Ans. 39.5% of the stroke.

HORSEPOWER OF COMPOUND ENGINES.

1307. The actual I. H. P. of a compound or triple-expansion engine may be obtained from the indicator cards. The method used is best shown by an example.

A triple-expansion engine has the volume of its cylinders in the ratio $1 : 2\frac{1}{2} : 6\frac{1}{4}$; that is, the low-pressure cylinder is $6\frac{1}{4}$ times as large as the high-pressure cylinder, and $\frac{6\frac{1}{4}}{2\frac{1}{2}} = 2\frac{1}{2}$ times as large as the intermediate cylinder. The low-pressure cylinder is 40″ × 40″. The engine makes 120 revolutions per minute. On measuring the actual cards it is found that the M. E. P. of the high-pressure cylinder is 80.5 pounds; the M. E. P. of the intermediate cylinder, 37.5 pounds, and of the low-pressure cylinder, 16.12 pounds. What is the I. H. P. of the engine?

It would be possible to calculate the I. H. P. by finding the work exerted by each cylinder separately, as if it were the cylinder of a simple engine, and then adding the works together. It is, however, easier to reduce all the pressures to the area of the low-pressure cylinder. This is done by dividing the M. E. P. of each cylinder by the ratio between its volume and the volume of the low-pressure cylinder. In the present case the M. E. P. of the high-pressure cylinder is 80.5. The volume of the low-pressure cylinder is 6.25

times that of the high-pressure cylinder, or, what is the same thing, the area of the low-pressure piston is $6\frac{1}{4}$ times that of the high-pressure piston. Therefore, to produce the same work, the M. E. P., when acting in the low-pressure cylinder, must be $\frac{1}{6.25}$ of what it was in the high-pressure cylinder. The M. E. P. of the small cylinder reduced to the area of the low-pressure cylinder is, therefore, $\frac{80.5}{6.25} = 12.88$ pounds. Likewise the M. E. P. of the intermediate, reduced to the low-pressure cylinder, is $\frac{37.5}{2.5} = 15$ pounds. The M. E. P. of the low pressure-cylinder, of course, remains the same. The total M. E. P., reduced to the low pressure-cylinder, is, therefore, $12.88 + 15 + 16.12 = 44$ pounds. Now, substituting this M. E. P., the area of the low-pressure cylinder, the length of stroke and revolutions per minute in formula **98**,

$$\text{I. H. P.} = \frac{PLAN}{33,000} = \frac{44 \times 40 \times 40^2 \times .7854 \times 120 \times 2}{12 \times 33,000} = 1,340.42.$$

1308. Fig. 299 shows an elevation of a **tandem compound non-condensing engine,** in which a is the high-

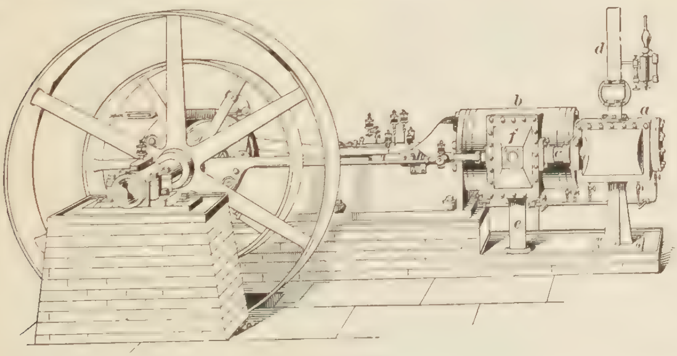


FIG. 299.

pressure cylinder, which is placed behind the low-pressure cylinder b. Many makers prefer to place the large cylinder behind, since it is then easier to remove the pistons and

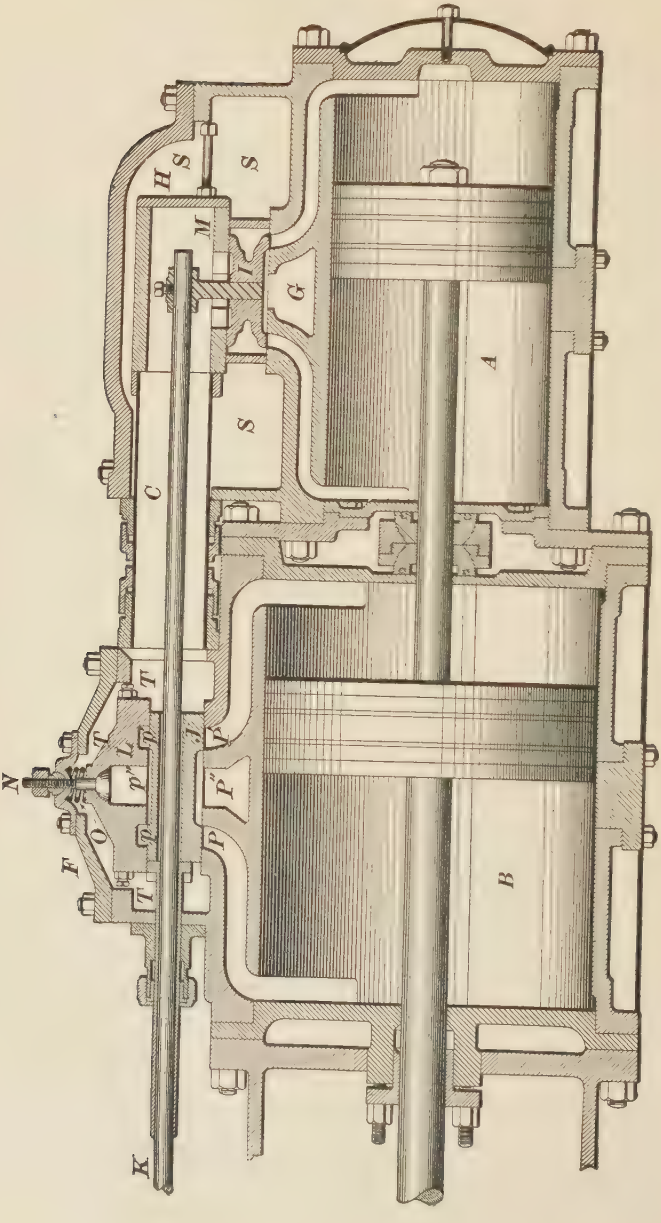


FIG. 300.

examine the cylinders in case of repairs. When the small cylinder is behind, it must be entirely removed from the engine before the pistons can be taken out, while, if the large cylinder is behind, the small piston can be pulled out. The steam pipe is shown at *d*, and the exhaust pipe at *e*. After the steam has expanded in *a*, it is discharged through the connecting pipe *c* into the steam chest *f* of the low-pressure cylinder. After doing its work in *b*, it is exhausted into the condenser or atmosphere through the pipe *e*. As will be seen, a shaft governor is used. Both pistons are, of course, attached to the same piston rod.

Fig. 300 shows a horizontal section taken through the center line of the cylinders of this engine. The live steam fills the space *S*, *S*, *S*, but has no communication with the pipe *C*, which leads to the steam chest *F* of the low-pressure cylinder, being prevented from entering *C* by the plate *H*. The exhaust port *G*, however, connects directly with *C* so that the steam, after expanding in *A*, exhausts through *G* into *C* and fills the space *T*, *T*, *T*, from whence it passes to the cylinder *B*. Both valves *I* and *J* are moved by the same valve stem *K*. Since the valves present many peculiarities over those before mentioned, a detailed description of them will now be given. The ordinary **D** slide-valve is subjected to a great pressure by the steam on its back. Thus, if, in a large engine, the valve is say 10″ long by 14″ wide and the boiler pressure is 100 lb., the pressure on the back of the valve when in its central position is $10 \times 14 \times 100 = 14{,}000$ pounds, or 7 tons. This great weight causes the valve seats to wear very fast, and, as they do not wear evenly, they soon leak badly. To obviate this difficulty, what are termed **balanced valves** are employed. Of this type are the valves shown in Fig. 300. Here flat plates *L* and *M* are placed on top of the valves. The plates and valves are ground together so that they fit perfectly, and no steam can get on the back of the valves. For convenience only the valve *J* will be described in connection with the balancing. The flat plate *L* is called a **pressure plate**, and is prevented from pressing the valve

with great force to its seat owing to the steam pressure on the back of the plate by reason of the bolt N. There must be some pressure upon the valve in order to keep it to its seat, and this is furnished by the spring O. The valve would still be unbalanced from beneath if the bottom of the pressure plate were a perfectly smooth surface; for, when in the position shown, the steam filling the ports P and P' presses upwards against the valve and forces it against the pressure plate. To counteract this, recesses p and p', having the same width and length as the steam ports P and P', are cut in the pressure plate and steam is allowed to enter each from the corresponding ports P and P', which are exactly opposite the recesses. For the same reason, the recess p'' balances the exhaust steam entering the port P''.

1309. By using balanced valves, another advantage is obtained besides the decreased wear; viz., the governor

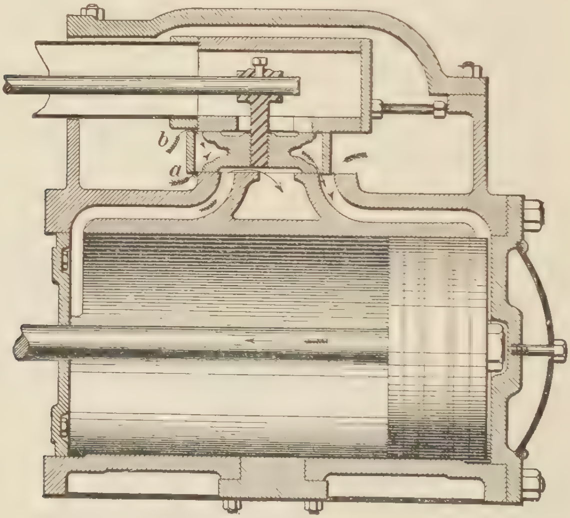


FIG. 301.

parts, eccentrics, etc., can be made much lighter. A pressure plate should not be used unless some provision has been made for allowing the valve to be raised from its seat in case water should be formed in the cylinder by the

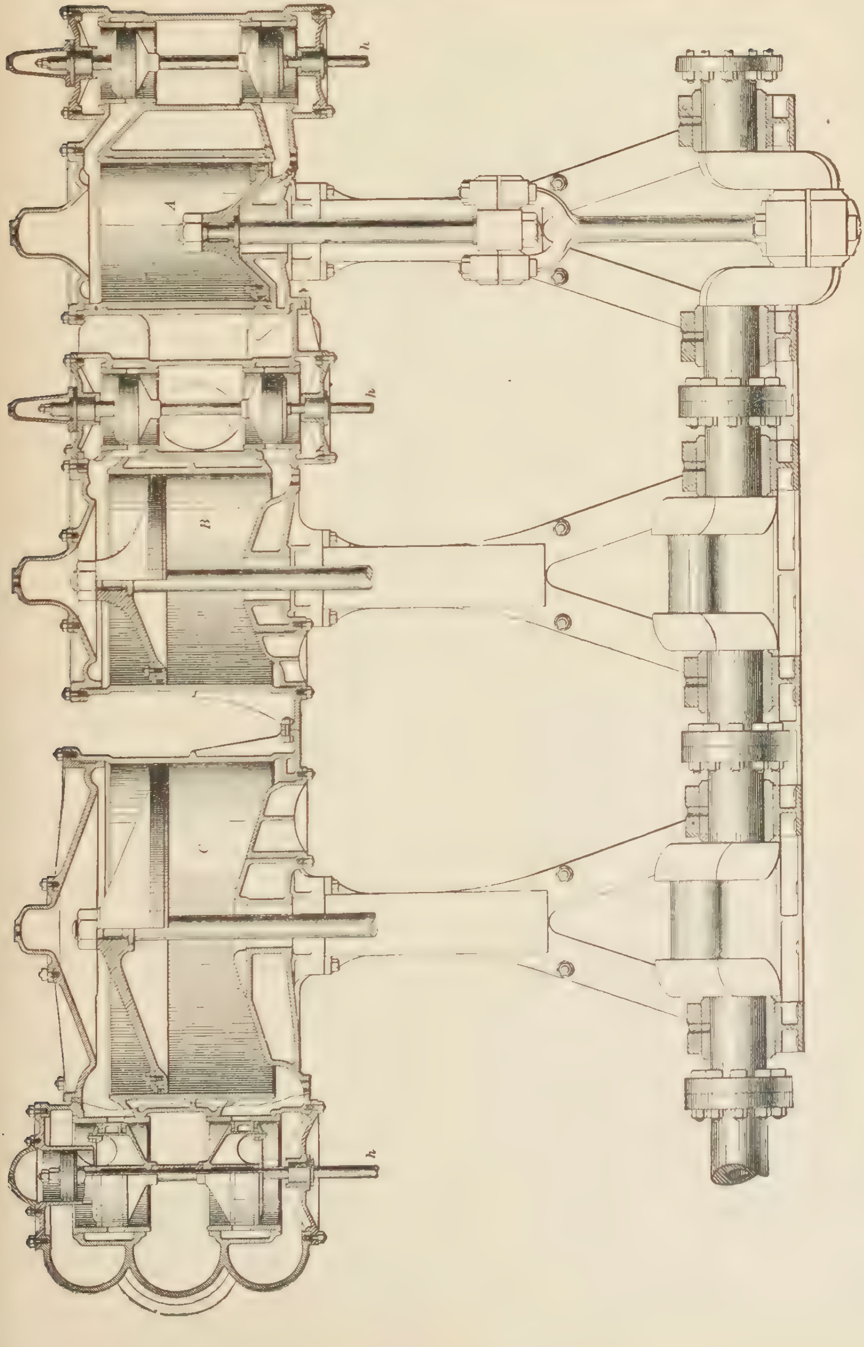


FIG. 302.

condensation of steam. In the present case, this is provided by the spring *O*, which will allow the pressure plate and valve to be raised together should water collect in the cylinder. Another peculiarity of the high-pressure cylinder valve is that it has three points or openings for the admission of steam to the cylinder. This is clearly shown in Fig. 301, which shows the valve and piston in position at the commencement of the stroke. The three points of admission are secured by making the valve hollow and allowing the steam to enter at *a* and *b*, flow through the valve and into the steam port, as shown by the arrows. The object of this is to allow a wider port opening without a corresponding increase in the travel of the valve.

Fig. 302 shows a section of a **triple-expansion marine engine.** Marine engines are always of the vertical type, since a vertical engine occupies much less floor space than a horizontal engine of the same capacity. *A* is the high-pressure, *B* the intermediate, and *C* the low-pressure cylinder. The cranks are placed 120° apart; hence, when the piston in *A* is beginning its stroke, the pistons in *B* and *C* are on the same level, one having traveled $\frac{2}{3}$ and the other $\frac{1}{3}$ of its stroke. The steam exhausts from *A* into *B*; from *B* into *C* and from *C* into the condenser. The valves (not shown in their proper positions) consist of pistons connected by a rod. They are moved by means of the stems *h* which are connected to eccentrics on the shaft *S* in the usual manner. A glance at the cut will show that these valves are perfectly balanced. The shaft *S* is made considerably larger than necessary for strength in order to obtain a greater bearing surface and thus reduce the amount of wear due to friction. It is also made hollow to reduce the weight.

The route of the steam through a triple engine may be more clearly seen in Fig. 303, which represents a plan and elevation of the cylinders of a marine engine (not the one of Fig. 302).

Steam from the boiler enters the two ends of the high-pressure steam chest *H*, through the pipe *A A′*. It is exhausted from the center of the steam chest through *B* to

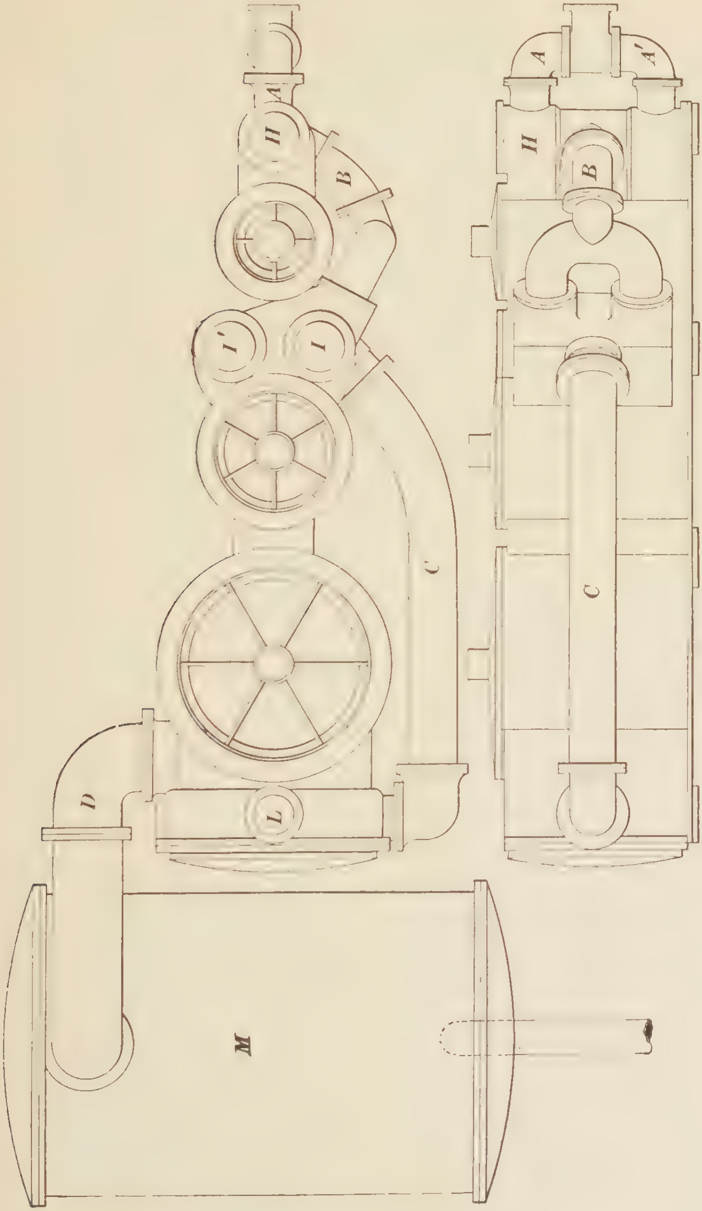


FIG. 303.

the two ends of the intermediate steam chest $I\ I'$. After expanding in the intermediate cylinder, it exhausts from the middle of I through the pipe C into the low-pressure steam chest L; and finally from L through the pipe D into the condenser M. After condensing in M, the steam, in the form of water, is pumped out of the condenser into the hot well, and from there into the boiler.

It may be observed that the steam chest I has two piston valves. In large triple-expansion engines, each cylinder may have two or even more piston valves.

MECHANICS OF THE STEAM ENGINE.

1310. In Fig. 304, let $O\ C$ represent the crank; then, the circle $C'\ C''\ M$ represents the path of the crank-pin.

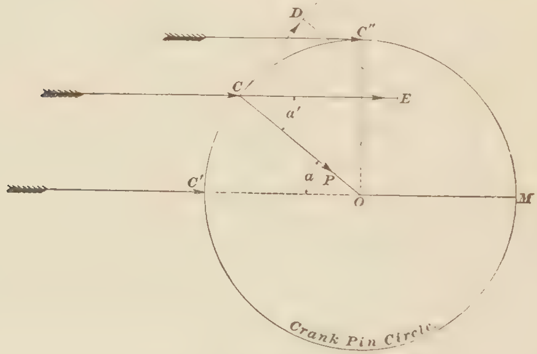


FIG. 304.

The steam exerts, through the piston, piston rod, and connecting-rod, a pressure on the crank-pin C. Let it be assumed that the connecting-rod is very long, so that the direction of the pressure on the pin is always horizontal.

Now, when the crank is in the position $O\ C'$, the horizontal pressure of the steam simply produces a pressure on the bearing of the crank-shaft; there is no tendency whatever to turn the crank around O as a center, and it is, therefore, said to be on the dead center. When the crank is in the

position $O\ C''$ *all* the pressure of the steam on the piston is expended in turning the crank around O as a center, and there is no pressure on the bearing. This is because the direction of the pressure at this point is at right angles to the crank, or, in other words, tangent to the crank circle.

When the crank is in some other position, as $O\ C$, there will be a tendency to turn the crank around O as a center, and also to produce a pressure on the crank-shaft bearing. To find the magnitudes of these forces tending to rotate the crank and tending to produce a pressure on the bearing, proceed as follows: Let $C\ E$ represent to some scale the horizontal pressure of force on the crank-pin at C. The turning force acts in the direction of the tangent $C\ D$, while the force which produces the pressure on the bearing acts along the crank, or in the direction $C\ O$. The force $C\ E$ may be resolved in the two directions $C\ D$ and $C\ O$ by means of the parallelogram of forces. From E, draw $E\ P$ parallel to $C\ D$ and $E\ D$ parallel to $C\ O$. Then, $C\ D$ is the tangential or turning force, and $C\ P$ the force producing pressure on the bearing, both to the same scale as $C\ E$.

Let a be the angle $C\ O\ C'$ which the crank makes with the horizontal. $C\ E$ and $C'\ O$ are parallel, and $C\ P$ and $C\ O$ are parallel (coincide); hence, by geometry, $C\ O\ C'$ and $P\ C\ E$ are equal, or angle $a =$ angle a'.

Tangential force $= C\ D = E\ P = C\ E \sin E\ C\ O = C\ E \sin a$, or *the force tending to turn the crank is equal to the horizontal force on the crank-pin multiplied by the sine of the angle which the crank makes with the horizontal.*

When the crank is at $C'\ O$, a is 0; therefore, $\sin a$ is 0 and the tangential force is 0, as it should be. When the crank is at $O\ C''$, a is 90°, $\sin a$ is 1, and the tangential force is the same as the horizontal force.

The radial force, or the force which produces pressure on the bearing, may be shown in the same manner to be equal to the horizontal force multiplied by the cosine of the angle which the crank makes with the horizontal, thus,

Radial force $= C\ P = C\ E \cos E\ C\ O = C\ E \cos a$.

EXAMPLE.—The horizontal force on the crank-pin is 6,000 pounds. What will be the tangential and radial forces when the crank makes an angle of 60° with horizontal direction of the force?

SOLUTION.—$a = 60°$; $\sin a = .866$; $\cos a = .5$.

Tangential force $= 6{,}000 \times \sin a = 6{,}000 \times .866 = 5{,}196$ lb.

Radial " $= 6{,}000 \times \cos a = 6{,}000 \times .5 = 3{,}000$ lb.

1311. A diagram showing the tangential pressure for every position of the crank may be easily constructed, as

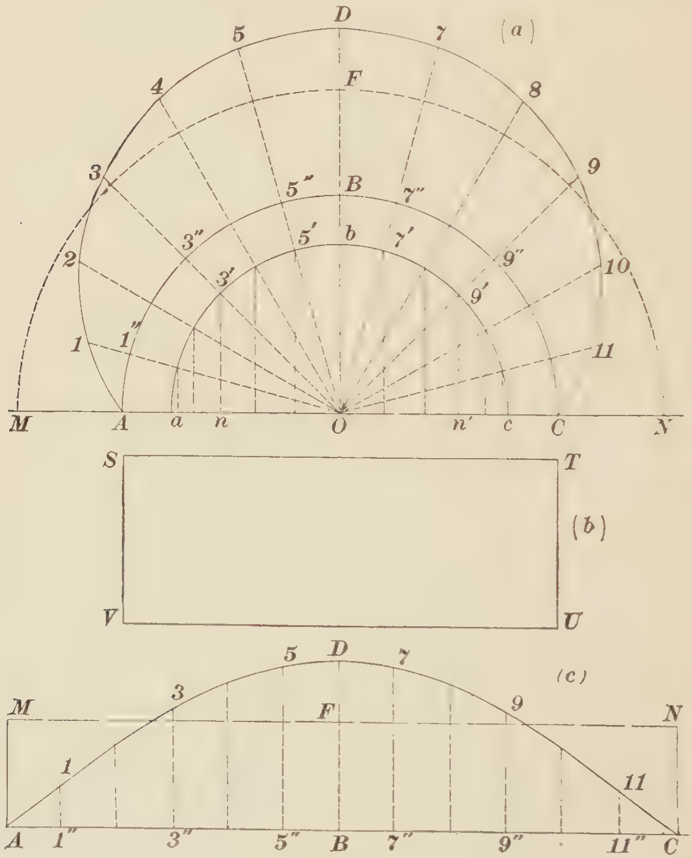


FIG. 305.

follows: For simplicity, assume the piston pressure to be constant throughout the stroke, as shown by the indicator

diagram or card, Fig. 305 (*b*); suppose, also, that the connecting rod is very long, so that the direction of pressure is always horizontal. Let the length of the crank be $O\ A$; then, with O as a center and $O\ A$ as a radius, describe one-half the crank-pin circle $A\ B\ C$. Let $O\ a$ represent the uniform pressure $s\ v$ and describe the semicircle $a\ b\ c$. Divide $a\ b\ c$ or $A\ B\ C$ into a convenient number of equal parts (12 is most convenient), and through each division draw the radial lines $O\ 1$, $O\ 2$, $O\ 3$, etc., prolonging them beyond $A\ B\ C$. From the points where these radial lines intersect the semicircle $a\ b\ c$ drop perpendiculars on line $a\ c$. Then, each perpendicular represents the tangential component of the pressure when the crank is in that position. For example, when the crank is at $O\ 3'$, the length $3'n$ represents the tangential pressure, since

$$3'n = O\ 3' \sin 3'O\ n = O\ a \sin 3'O\ n = \text{horizontal pressure} \times \text{sine of crank angle.}$$

Now, lay off these perpendiculars, each on its own radial line outwards from the crank-pin circle $A\ B\ C$; that is, lay off $3'n$ on the radial line $O\ 3$, the length $3\text{-}3''$ being made equal to the length $3'n$.

We thus obtain the series of points *1*, *2*, *3*, etc.; the curve $A\ D\ C$, drawn through these points, will represent the tangential pressures for all points of the stroke.

It will be noticed that at A and C, the dead points, the tangential pressure is 0, and at D it is equal to the horizontal pressure.

In Fig. 305 (*c*), the tangential pressure diagram is represented with a straight base. The semicircle $A\ B\ C$ has been straightened out, the ordinates $1\text{-}1''$, $2\text{-}2''$, etc., remaining the same as before.

Since the ordinates of (*c*) represent the tangential pressures on the crank to the same scale that the ordinates of (*b*) represent pressures on the piston, and since the length $A\ C$ represents the distance passed through by the crank to the same scale that $V\ U$ represents the distance passed through by the piston, it follows that the area of (*c*)

represents the work done by the crank-pin during a half revolution of the crank, or during one stroke of the piston.

The work done on the piston by the steam must be equal to the work given up by the crank-pin. Therefore, since (*b*) and (*c*) have the same scale of pressures and distances, since (*b*) represents the work done on the piston and (*c*) represents the work done by the crank-pin, their areas must be equal; and so they will be found to be by actual measurement.

The "mean ordinate" of (*c*) may now be found from the above considerations. The length of the semicircle $A\ B\ C$ of (*a*) is $\frac{\pi}{2}$ times the length of the diameter $A\ C$; but the base $A\ B\ C$ of the diagram (*c*) is equal in length to the semicircle $A\ B\ C$, and the length $V\ U$ of (*b*) is equal to the diameter $A\ C$, since both represent the length of stroke. Therefore, the base $A\ B\ C$ of (*c*) is $\frac{\pi}{2}$ times as long as the base $U\ V$ of (*b*). The area of (*b*) and (*c*) are the same; consequently, the mean ordinate of (*b*) must be $\frac{\pi}{2}$ times that of (*c*), or $S\ V = \frac{\pi}{2} A\ M$; therefore, $A\ M = \frac{2\ S\ V}{\pi}$.

The mean ordinate of the diagram of tangential pressures on the crank-pin is always $\frac{2}{\pi}$ times the mean ordinate or M. E. P. of the indicator card.

In both (*a*) and (*c*), $M\ F\ N$ is the line of average tangential pressures, and in both cases is drawn parallel to $A\ B\ C$ at a distance from it equal to the mean ordinate $A\ M$.

The case just described is much simpler than is ever met with in practice. The pressure of the steam is rarely constant throughout the whole stroke, and the connecting rod is never so long that the pressure exerted by it on the crank pin may be regarded as always horizontal.

1312. In order to apply the foregoing principles to an actual case, it is necessary to use the net pressure (forward

pressure, less the back pressure on the other side of the piston) on the piston during one-half a revolution. This is easily done by combining the two curves *A B C D* and *K L C M*, Fig. 267 (which represent the steam pressure on opposite sides of the piston during the forward stroke), as shown in Fig. 306, where *A B a c* represents *A B C D*, and *c f a b* represents *K L C M*. Joining *A* and *e* and *b* and *c* by straight lines, the figure is completed. Any ordinate, as *g h*, measured to the scale of the indicator spring, is the net pressure on the piston urging it ahead when it occupies the position of *h* of its stroke. The net pressure is 0 at *a*,

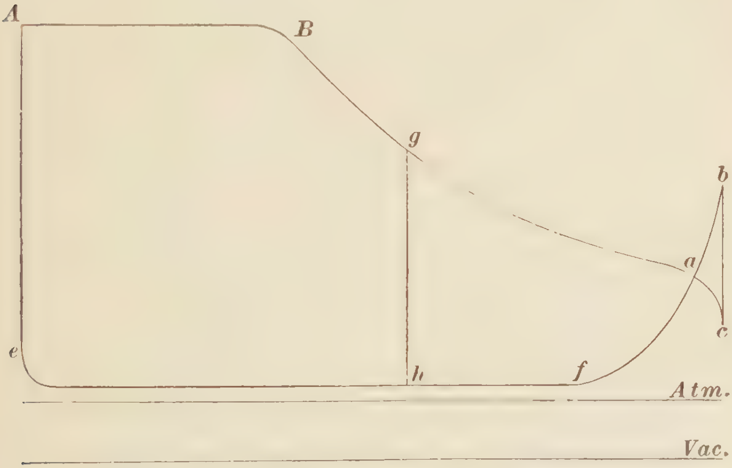


FIG. 306.

i. e., the pressure is the same on both sides of the piston. Between *a* and *b c* the net pressure is negative, or, in other words, the back pressure is greater than the forward pressure, and the piston is carried to the end of its stroke solely by the energy stored in the fly-wheel.

1313. Let us now take a diagram of *net* pressures on the piston like the one in the figure just shown, and, assuming the connecting-rod to be four times the length of the crank, work out the corresponding diagram of tangential pressures.

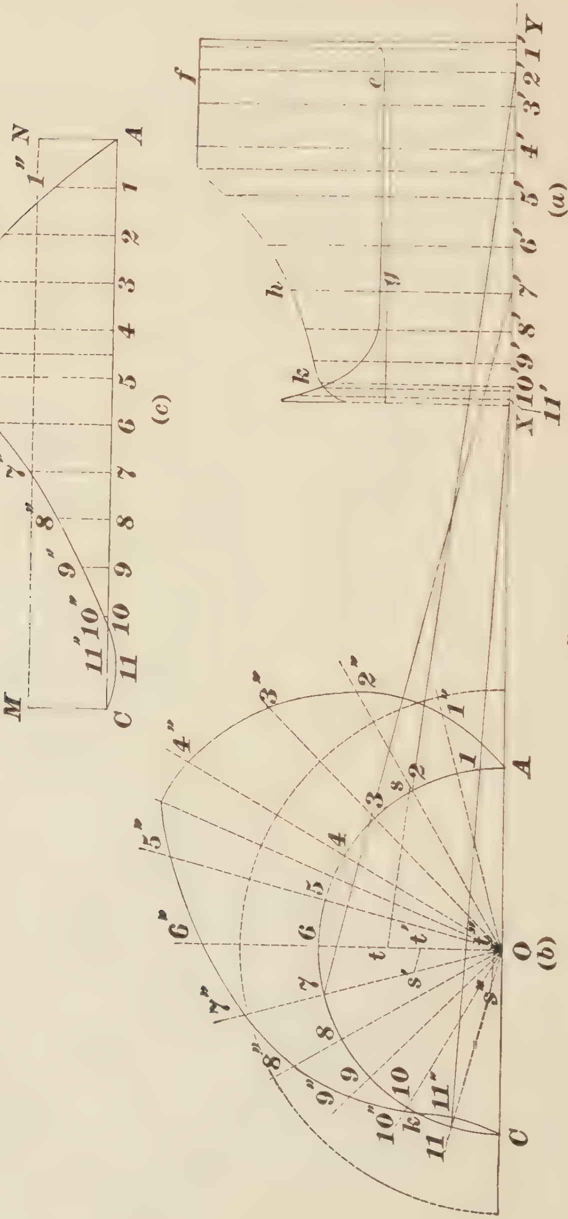


FIG. 307.

The diagram or card is shown at (*a*) Fig. 307. At any convenient distance below the card, draw a line $C\ Y$ parallel to the atmospheric line of the cards. Project the two ends of the card down on $C\ Y$, locating the points X and Y. Lay off, on the line $C\ Y$, $Y\ A = X\ C = 2\ X\ Y = 4\ O\ A =$ the length of the connecting-rod. Take the point O midway between A and C, and with $O\ A$ as a radius, strike the semicircle $A\ 6\ C$. $X\ Y$ now represents the length of the stroke to the scale of the diagram, and $A\ 6\ C$ the path of the crank-pin. Divide $A\ 6\ C$ into a convenient number of equal parts, and through the points of division draw the radial lines $O\ 1$, $O\ 2$, $O\ 3$, etc., prolonging them some distance beyond the semicircle. Now find the positions of the piston corresponding to the crank positions $O\ 1$, $O\ 2$, etc. This may be done by taking the length $A\ Y$ of the connecting-rod with a pair of dividers; then, placing one leg of the dividers on 1, 2, 3, etc., in succession, strike off the piston positions $1'$, $2'$, $3'$ on the line $X\ Y$. The tangential pressure may be found in the following manner: Suppose it is desired to find the tangential pressure on the crank when it is in the position $O\ 2$. The pressure on the piston at this point is shown on the card to be $e\ f$. Lay off from O the length $e\ f$ on the radial line $O\ 2$, thereby obtaining the length $O\ s$. Draw in the connecting-rod $22'$, and through s, draw $s\ t$ parallel to $22'$ and intersecting the vertical line $O\ 6''\ t$. Then, $O\ t$ is the tangential pressure to the same scale that $e\ f = O\ s$ is the pressure on the piston. When the crank is at 7 the piston is at $7'$, and the pressure on it from the card is $g\ h$. $O\ s'$ is laid off on $O\ 7$ equal to $g\ h$, $s'\ t'$ is drawn parallel to $7\ 7'$; then, $O\ t'$ is the tangential pressure on the crank when it is in the position $O\ 7$. In this manner, the tangential pressure can be found for all positions of the crank, and, being laid off radially from the crank-pin circle, as in the previous case, they form the curve $A\ 4''\ 7''\ k\ C$. At k, the pressures on both sides of the piston are equal, and, consequently, the tangential pressure is zero. The point k in (*b*) is determined by dropping a perpendicular from k to $X\ Y$, and with the point of intersection as a center and a

radius equal to the length of the connecting-rod, describe an arc cutting *C 6 A* in *k*. Beyond *k*, the back pressure exceeds the forward pressure, and the tangential pressure must be laid off below the semicircle *C k 6 A*, as shown. As before, the semicircle *A 6 C* may be laid out straight, and the diagram shown in (*c*) obtained. The area of the latter will again be found to be equal to the area of the diagram (*a*). The line *M N*, as before, represents the average tangential pressures on the crank-pin. It is usually a good plan to find the crank and piston positions at cut-off and get the tangential pressure at that point, since a sharp change in the curve will there occur, as shown in the figure.

1314. The load on the engine, or the resistance against which the engine works, is nearly always constant. That is, it requires a practically constant force to drive the shafting or machines. It is, therefore, very desirable that the

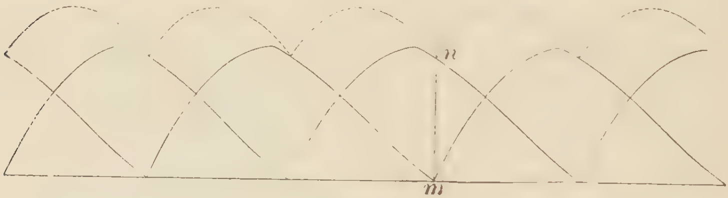


FIG. 308.

force turning the crank-shaft should be as nearly constant as possible. It has been shown by the diagrams of Figs. 305 and 307 that this tangential force is very far from being constant; that at the dead points it is zero and near the middle of the stroke it is greatest. It can now be shown why compound and duplex engines, which have their cranks at right angles, possess advantages over the simple engine. In Fig. 308 is shown a diagram for a cross-compound engine with cranks at right angles. When one crank is beginning its stroke, the other is at the middle of the stroke, and vice versa. Therefore, when the curve of one crank touches the base line, as at *m*, the other will be at or near its highest position, as at *n*. Now, adding the corresponding ordinates of the two curves together, we obtain the dotted curve

which, therefore, represents the total tangential pressure tending to turn the crank-shaft. It is apparent at a glance how much more nearly constant is the tangential pressure of the compound engine diagram as compared with the tangential pressure of the simple engine diagram. Consequently, with the same steam pressure and weight of fly-wheel, the compound or duplex engine with crank at 90° will run more steadily than the simple engine, while a triple-expansion engine with 3 cranks, making angles of 120° with each other, will run steadier than the compound.

1315. A tandem compound engine also has a mechanical advantage over the simple engine, as a little consideration will show. It was proven in Art. **1301** that a compound engine was equivalent to a simple engine whose cylinder was of the same dimensions as the low-pressure cylinder, and which expanded its steam the same number of times as the compound. For convenience of illustration, assume the cut-off in the high-pressure cylinder to be $\frac{1}{2}$, and that the ratio of the volumes of the two cylinders is 1 : 4; that there is no clearance and no receiver; that there is no loss of pressure due to wire-drawing, friction, etc., and, finally, that the steam is carried full stroke in the low-pressure cylinder. The total number of expansions is 8, 2 in the small cylinder and 4 in the large cylinder. Assume the steam pressure to be 120 pounds, absolute, and the absolute back pressure to be 15 pounds. Denote the area of the large piston by A; then, the area of the small piston is $\frac{A}{4}$. The terminal pressure in the small cylinder is, evidently, 60 pounds, and this is the initial forward pressure in the large cylinder and the initial back pressure in the small cylinder.

First consider the simple engine. The initial forward pressure is 120 pounds and the back pressure 15 pounds; hence, the net pressure urging the piston ahead is $120 - 15 = 105$ pounds. The total force acting on the piston is $105\,A$.

Considering the compound engine, the initial forward

pressure in the small cylinder is 120 pounds; the back pressure 60 pounds, and the net pressure $120 - 60 = 60$ pounds. Since the area of the small piston is $\frac{A}{4}$, the total force tending to drive the small piston forward is $60 \times \frac{A}{4} = 15\ A$. The initial forward pressure in the low-pressure cylinder is 60 pounds, the back pressure is 15 pounds, and the net pressure urging the piston ahead is $60 - 15 = 45$ pounds. The total force tending to drive the large piston forward is $45\ A$. The total initial force acting on both pistons of the compound is $45\ A + 15\ A = 60\ A$; in the simple engine it was $105\ A$. Since the greatest strains occur when the forces which produce them are greatest, it is evident, from the above, that the various parts of a compound engine (connecting-rod, crank, shaft, etc.) will not need to be made so large as in a simple engine having the same mean effective pressure, and the volume of whose cylinder equals the volume of the low-pressure cylinder. This is, however, a disadvantage when the engine has a very heavy load to start, as in the case of a locomotive, for in that case the compound engine might not be able to start, although it could keep itself in motion after it had once been started.

CONDENSERS.

1316. It has been shown that the thermal efficiency of the steam engine, $\frac{T_1 - T_2}{T_1}$, may be increased by either raising the temperature, T_1, of the live steam, or by lowering the temperature, T_2, of the exhaust steam. T_1 may be raised by increasing the boiler pressure of the steam; T_2 may be lowered by using a **condenser.**

In non-condensing engines—that is, engines which are not supplied with a condenser—the steam is exhausted into the atmosphere, and therefore the exhaust steam must have, at least, the pressure of the atmosphere; in practice the

back pressure of steam in a non-condensing engine is scarcely ever less than 16 pounds above vacuum, and is oftener 17 pounds or more. In good condensing engines the back pressure is often as low as 2 pounds above vacuum.

1317. Suppose that the boiler pressure of the steam is 80 pounds absolute, the temperature corresponding to the pressure is, from the steam table, 311.9° F., and the absolute temperature is, therefore, $460° + 311.9° = 771.9°$ F. The absolute temperature corresponding to a pressure of 17 pounds is $460° + 219.5° = 679.5°$ F., and corresponding to a pressure of 3 pounds is $460° + 141.7° = 601.7°$ F. The thermal efficiency of the engine, if non-condensing, is $\frac{T_1 - T_2}{T_1} = \frac{771.9 - 679.5}{771.9} = 12\%$, nearly; if condensing to 3 pounds absolute, the efficiency is $\frac{T_1 - T_2}{T_1} = \frac{771.9 - 601.7}{771.9} = 22\%$.

1318. The increase of economy by the use of the condenser may be shown in another manner. Let $A\,B\,C\,D\,E\,F$, Fig. 309, be an indicator card from a non-condensing engine. $M\,N$ is the atmospheric line and $O\,X$ the vacuum line. The

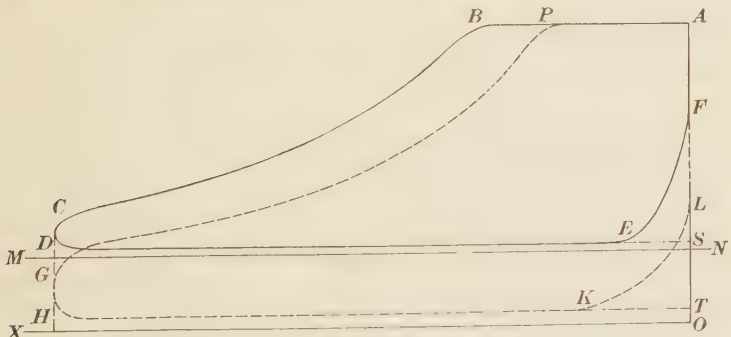


FIG. 309.

back pressure, as shown by the card, is $O\,S$. The area of the card represents to some scale the work done per stroke. Now let a condenser be attached to the engine. The back pressure will be lowered to $O\,T$, the line $H\,K$, instead of

D E, now being the lower line of the card, and *A B C H K L* will be the new card, its area, as before, representing the work done per stroke. Hence, by adding a condenser to the engine, the work per stroke has been increased by an amount represented by the area *F E D H K L*, the steam consumption remaining the same. Suppose the steam to be cut off at a point *P*, making the area of the card, *A P G H K L*, equal to the area of the original card, *A B C D E F*. Then, the work per stroke is the same in both engines, but the condensing engine uses an amount of steam per stroke represented by the length *A P*, while the non-condensing engine uses an amount represented by *A B*. Either case shows the economy of the condenser.

1319. There are two types of condensers in general use: The **surface condenser** and the **jet condenser.**

In the former, the exhaust steam comes in contact with a large area of metallic surface, which is kept cool by contact with cold water. In the latter, the exhaust steam, on entering the condenser, comes in contact with a jet of cold water. In either case, the entering steam is condensed to water, and, in consequence, a partial vacuum is formed. If a sufficient amount of cold water was used, the steam on entering would instantly condense, and a practically perfect vacuum would be obtained, were it not for the fact that the feed water of the boiler always contains a small quantity of air, which passes with the exhaust steam into the condenser, and therefore partially destroys the vacuum. To get rid of this air, the condenser is fitted with an air pump, which pumps out both the air and the water, into which the steam condenses.

1320. The Surface Condenser.—Fig. 310 is a perspective view of the Wheeler surface condenser. Fig. 311 is a sectional view of the same. The cold condensing water is drawn from some water supply through *M*, and forced by the circulating pump *Q* into the inlet *C* of the condenser. From *C* the water is forced to the chamber *F*, and flows, as indicated by the arrows, through the inner tubes of the

lower layer of double tubing to the left, and having passed through their entire length, it returns through the space

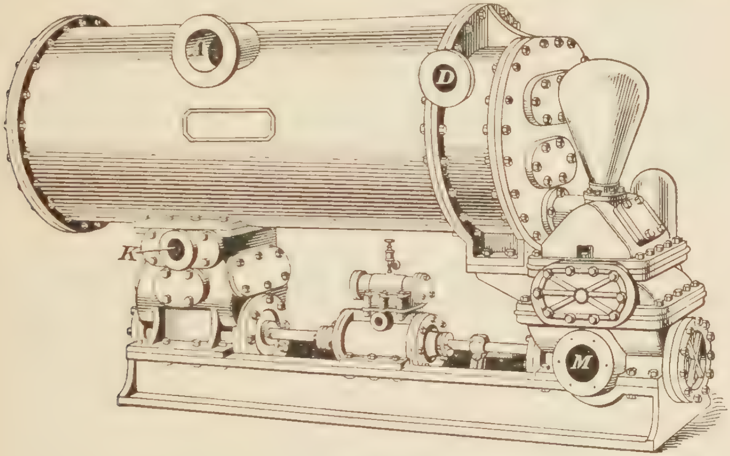


FIG. 310.

between the outside of the inner and inside of the outer tubes into the chamber *G*. Fig. 312 shows more clearly the arrangement of this "double tubing." From *G* (Fig. 311) it passes through *E* to *H*, and from *H* to *I* through the

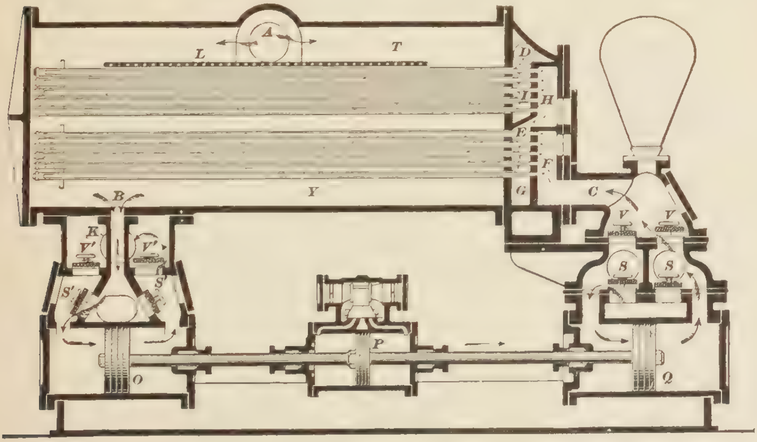


FIG. 311.

upper layer of double tubing, as has already been explained. From *I*, it is discharged through the nozzle *D*, carrying with

it all the heat it has received by coming in contact with the two layers of double tubing.

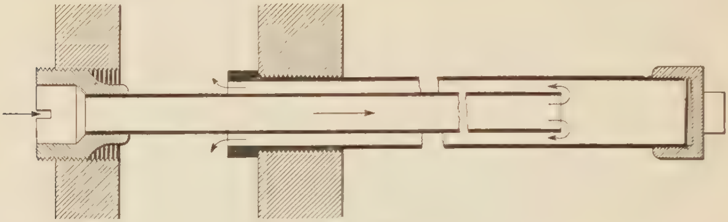


FIG. 312.

The nozzle at A is connected with the exhaust pipe of the steam cylinder of an engine. The movement of the air pump piston O draws air through the orifice B from the condenser cylinder and discharges it through the valves and nozzle K in a manner clearly indicated by the arrows. The valves S' and V' are opened and closed automatically by the pressure of the air beneath them and by the pressure of the air and springs above them. A partial vacuum is created in the condenser cylinder Y by the action of the air pump, thus sucking the condensed steam from the engine cylinder into the condenser cylinder.

As the exhaust steam enters the condenser cylinder through A, it first comes in contact with the perforated scattering plate L, which protects the upper tubing from the damaging effect of direct contact with the exhaust steam. The steam comes in contact with the cold tubes, through which the cold water is being pumped, and condenses. As soon as this occurs, the condensed exhaust steam collects at the bottom of the condenser cylinder and runs through B into the air pump cylinder, from which it is discharged while still heated and made use of as boiler feed-water. The temperature of this condensed exhaust steam boiler feed-water will be less as the vacuum in the condenser cylinder is more complete. The advantage of making use of this condensed exhaust steam as boiler feed-water will be clearly seen when it is remembered that it has a higher temperature than the ordinary feed-water, and will, therefore, require less fuel in converting it into steam.

In this condenser, the circulating and air pumps are run by an independent steam cylinder *P*. They are often connected directly to the main engine, and have their motion imparted to them by some of its moving parts, generally the crank-shaft.

1321. The Jet Condenser.—In Fig. 313 is shown a section of a Worthington independent jet condenser. The

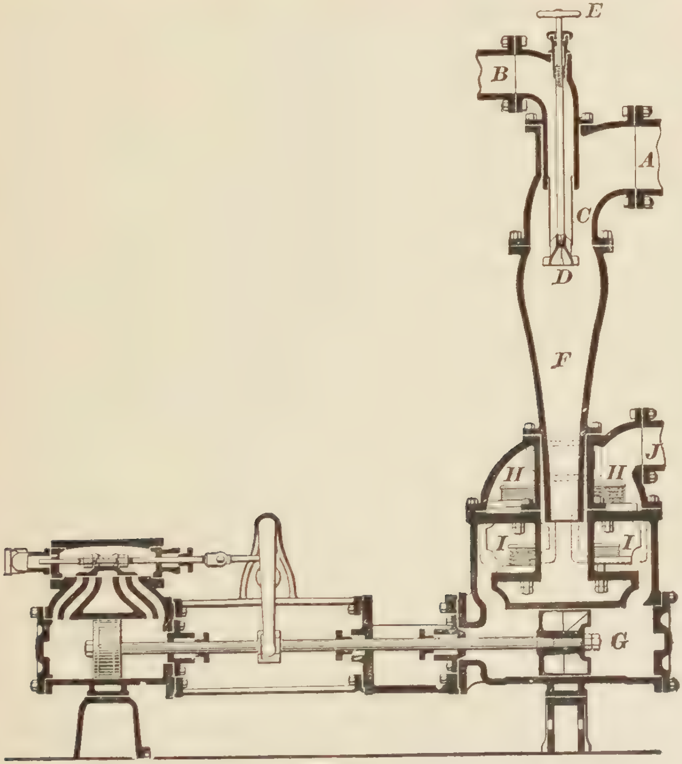


FIG. 313.

cold water enters the condenser at *B*, passes down the spray pipe *C*, and is broken into a fine spray by the cone *D*. The exhaust steam in the meantime comes in at *A*, and, mingling with the spray of cold water, is rapidly condensed. The velocity of the entering steam is imparted to the water, and

the whole mixture of steam, water, uncondensed vapor, and air is carried with a high velocity through the cone *F* into the pump cylinder *G*, whence it is forced by the pump through the discharge pipe *J*.

1322. In Fig. 314 is shown a jet condenser in connection with the boiler and engine. The exhaust pipe *A* leads

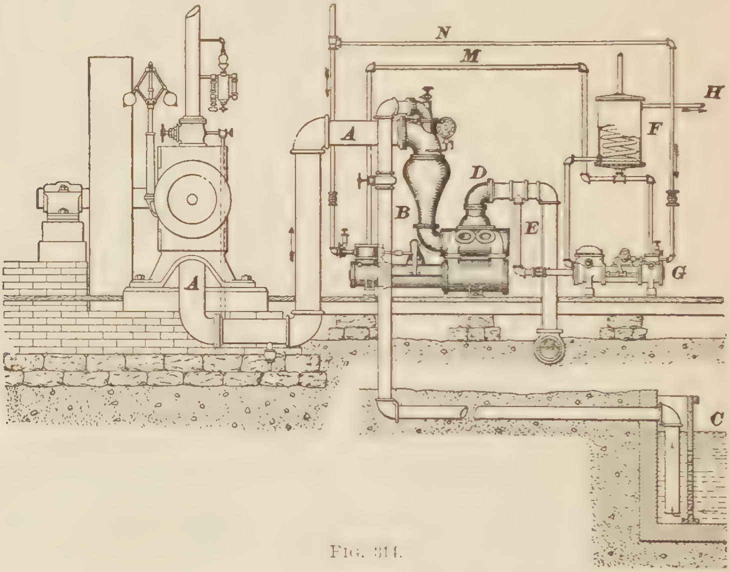


FIG. 314.

directly to the condenser. The injection pipe *B* draws water from the reservoir *C*. After the steam is condensed, the mixture of exhaust steam and injection water is discharged through *D* into the sewer. A portion of this discharge, however, flows through *E* to the feed pump *G*, which forces it through the coil in the heater *F* to the pipe *H* leading to the boiler. The exhaust from the two pumps is discharged into the feed-water heater through the pipe *M*. It will be noticed that water from the overflow pipe *D* enters the feed pump under a slight head. This is because the water is heated by the exhaust steam, and hot water can not be raised by a pump like cold water. *N* is a pipe leading from the boiler, and supplies steam for both pumps.

1323. The **water required by a condenser** may be calculated as follows:

Let t_1 = the temperature of departing condensing water;
t_2 = " " entering " "
t_3 = " " the condensed steam upon leaving the condenser;
H = total heat of one pound of steam at the pressure of the exhaust;
W = the weight of water required per pound of steam condensed.

The heat given up by the one pound of steam condensing to the temperature t_3 is $H - (t_3 - 32°)$ B. T. U.; the heat absorbed by the water is $W(t_1 - t_2)$. Therefore,

$$W(t_1 - t_2) = (H - t_3 + 32), \text{ or } W = \frac{H - t_3 + 32}{t_1 - t_2}. \quad \textbf{(105.)}$$

EXAMPLE.—The steam exhausts into the condenser at a pressure of 4 pounds absolute. The temperature of the condensing water on entering is 60°, and on leaving 100°. The temperature of the condensed steam on entering the air pump is 140°. How many pounds of condensing water are required per pound of steam?

SOLUTION.—From the steam table, the total heat of a pound of steam at 4 pounds pressure above vacuum is 1,128.641 B. T. U.

Then, $W = \frac{H - t_3 + 32}{t_1 - t_2} = \frac{1{,}128.641 - 140 + 32}{100 - 60} = 25.16$ pounds. Ans.

In the jet condenser the final temperatures of the cooling water and the condensed steam are the same—that is, $t_3 = t_1$.

EXAMPLE.—Steam exhausts into a jet condenser at a pressure of two pounds above vacuum. The temperature of the condensing water is 60°, and the temperature of the mixture as it enters the pump is 135°. How much water must be used per pound of steam?

$$W = \frac{H - t_3 + 32}{t_1 - t_2} = \frac{1{,}120.462 - 135 + 32}{135 - 60} = 13.566 \text{ pounds. Ans.}$$

1324. The surface condenser is more generally used than the jet condenser, especially in marine practice. They cost more originally and require more condensing water, but they possess the great advantage of allowing only the condensed steam to return to the boiler; hence, any water, no matter how impure, may be used for condensing. The jet

condenser, on the other hand, pumps both condensing water and condensed steam into the hot well, and hence, if impure water be used, it will find its way into the boiler and form a scale.

1325. Cooling Surface.—The cooling surface required by a surface condenser may be calculated as follows:

Let S = the required surface in square feet;

W = total weight of steam used per hour;

$$\text{Then, } S = .0944\,W. \quad \textbf{(106.)}$$

EXAMPLE.—What cooling surface should be given the surface condenser of an engine developing 240 I. H. P. and using 23 pounds of steam per I. H. P. per hour?

SOLUTION.—Using formula **106,**

$$S = .0944\,W = .0944 \times 240 \times 23 = 521.1 \text{ sq. ft. Ans.}$$

EXAMPLES FOR PRACTICE.

1. How many gallons of water are needed per hour to condense the steam from a 150 horsepower engine which uses 31.63 pounds of steam per horsepower per hour? The pressure of the exhaust is 4 pounds absolute; the temperature of the condensed steam is 172°; of the water, as it enters the condenser, 66°, and on leaving, 112°.

Ans. 12,204.5 gal. per hour.

2. A jet condenser is attached to a direct-acting steam pump which runs at a piston speed of 111 feet per minute. The steam cylinder is 28″ × 36″ and the steam pressure is 70 pounds gauge. How much water must be furnished per hour to the condenser, if its temperature on entering is 54° and on leaving 160°, the vacuum gauge showing 20″?

Ans. 6,352.81 gal. per hour.

NOTE.—The steam is never cut off in a simple, direct-acting pump.

3. What cooling surface is necessary for an engine developing 360 I. H. P. and using 28 pounds of steam per I. H. P. per hour?

Ans. 951.552 sq. ft.

FLY-WHEELS.

1326. The office of the *fly-wheel* is somewhat similar to that of the governor, since each is used to obtain regularity of speed. It is the duty of the governor to adjust the effort of the engine to any large or permanent variation of the

load, such as would be caused by throwing the machinery in or out of gear. It is the duty of the fly-wheel, on the other hand, to adjust the effort of the engine to sudden fluctuations of the resistance, which may occur during a single stroke of the engine. It is also the duty of the fly-wheel to equalize the varying tangential effort on the crank pin by storing energy while the piston is in the middle of the stroke, where the crank effort is greater than the resistance, and restoring it when the crank is at the dead point, where the tangential effort is zero.

1327. In Fig. 315, let $A\ C\ B$ represent the tangential pressures on the crank-pin during one stroke of the piston. $A\ B$ is the length of the semi-circumference of the crank-pin circle; $A\ M$ is the mean ordinate, and $A\ B\ N\ M$ represents the constant resistance to the turning of the crank. The effort and resistance must be equal; therefore, area $A\ C\ B$ = area $A\ B\ N\ M$. It follows, then, that area $C\ D\ E$ = area $A\ M\ D$ + area $B\ N\ E$. At A the tangential effort is zero, since the crank is at a dead point. From A to S the tangential effort remains less than the resistance. The work done on the crank during that period is represented by the area $A\ D\ S$, while the resistance is represented by the area $A\ M\ D\ S$. Hence, the resistance is greater than the effort by the area $A\ M\ D$. At S the effort and resistance are equal. From S to T the effort is greater than the resistance by an amount represented by the area $D\ C\ E$. At T the effort is again equal to the resistance, and for the remainder of the stroke it again becomes less than the resistance.

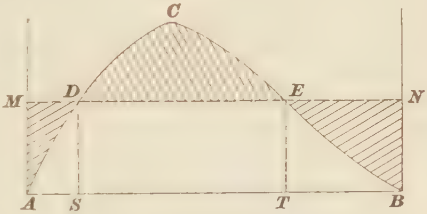


FIG. 315.

From A to S the work done by the steam was shown to be less than the resistance to be overcome; therefore, the work represented by the area $A\ M\ D$ must have been

obtained from some of the moving parts of the engine. The kinetic energy of a moving body is $\frac{W v^2}{2 g}$. In order to give up energy, the v of the above expression must be diminished. That is what actually takes place. All of the moving parts of the engine, the reciprocating parts, shaft, and fly-wheel, slow down a little, and, in so doing, give up enough of their kinetic energy to overcome the resistance and carry the engine past the dead center. From S to T the effort is greater than the resistance, and, consequently, the surplus energy represented by the area $C D E$ is stored up in the moving parts—that is, their velocity is increased during the part of the stroke in question, and with it their kinetic energy. For the remainder of the stroke the moving parts again slow down and give up enough energy to overcome the excess resistance $B N E$.

1328. The **weight of the fly-wheel** may be found in the following manner:

Let V_1 = the greatest velocity of the crank-pin in feet per second;

V_2 = the least velocity of the crank-pin in feet per second;

V_0 = average velocity of crank-pin in feet per second;

W = required weight of fly-wheel in pounds;

H = the number of foot-pounds per square inch of piston represented by the area $C D E$;

A = area of piston in square inches;

n = ratio between radius of fly-wheel and length of crank;

$E = \frac{V_1 - V_2}{V_0}$ = coefficient of unsteadiness.

The average velocity V_0 is known, since it is $\frac{\pi}{2}$ times the piston speed in feet per second. V_1 and V_2 are assumed so that the fraction $\frac{V_1 - V_2}{V_0} = E$ shall not exceed a certain value.

The following values of E agree well with ordinary practice:

Pumping engines,	$\frac{1}{20}$.
Engines driving machine tools,	$\frac{1}{35}$.
" " textile machinery,	$\frac{1}{40}$.
" " spinning machinery,	$\frac{1}{50}$ to $\frac{1}{100}$.
" " electric machinery,	$\frac{1}{125}$ to $\frac{1}{150}$

H can be found directly from the diagram by multiplying the area $C\ D\ E$ in square inches by the vertical scale of pressures and the horizontal scale of distances. A is, of course, known, and n may be assumed at pleasure.

At the point D the crank-pin has the velocity V_2; since the radius of the fly-wheel rim is n times the length of crank, the velocity of the rim at D must be $n\ V_2$. Likewise, at E the crank-pin will have its greatest velocity V_1, and the fly-wheel rim the velocity $n\ V_1$.

The kinetic energy of the fly-wheel rim at D is $\dfrac{W(n V_2)^2}{2g}$; and at E it is $\dfrac{W(n V_1)^2}{2g}$. The kinetic energy stored up in passing from D to E is, therefore, $\dfrac{W n^2 V_1^2}{2g} - \dfrac{W n^2 V_2^2}{2g} = \dfrac{W n^2}{g}\left(\dfrac{V_1^2 - V_2^2}{2}\right)$ foot-pounds. But this kinetic energy stored up is equal to the work represented by the area $C D E$. Hence,

$$\frac{W n^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right) = A \times H. \quad (a).$$

But $\dfrac{V_1 + V_2}{2} = V_0$, the average velocity,

and $\dfrac{V_1 - V_2}{V_0} = E;$

therefore, $\left(\dfrac{V_1 - V_2}{V_0}\right)\left(\dfrac{V_1 + V_2}{2}\right) = \dfrac{V_1^2 - V_2^2}{2 V_0} = E V_0,$

or $\dfrac{V_1^2 - V_2^2}{2} = E V_0^2.$

Substituting this in (a),

$$\frac{W n^2}{g}\left(\frac{V_1^2 - V_2^2}{2}\right) = \frac{W n^2 E V_0^2}{g} = A H.$$

Whence, $W = \dfrac{A H g}{n^2 E V_0^2}$. **(107.)**

EXAMPLE.—Given a single cylinder engine making 100 revolutions per minute, with a stroke of 3 feet and a cylinder diameter of 20 inches. A diagram similar to Fig. 315 is drawn, and it is found that the area of the portion $C\,D\,E$ is .86 sq. in. The diagram was drawn so that a vertical height of one inch represents 40 pounds pressure per square inch, and a horizontal distance of one inch represents a crank-pin travel of $1\frac{1}{2}$ feet. Assume coefficient of unsteadiness E to be $\frac{1}{40}$, and the ratio n as 4. Find the necessary weight of the fly-wheel rim.

SOLUTION.—Piston speed $= \frac{2 \times 3 \times 100}{60} = 10$ ft. per sec. Average speed of crank-pin = speed of piston $\times \frac{\pi}{2} = 10 \times \frac{3.1416}{2} = 15.7$ ft. per sec. $= V_0$. Area of piston $= .7854 \times 20^2 = 314.16$ sq. in. Work per square inch of piston represented by area $C\,D\,E = .86 \times 40 \times 1\frac{1}{2} = 51.6$ ft.-lb. $= H$. $W = \frac{A\,H\,g}{n^2 E V_0^2} = \frac{314.16 \times 51.6 \times 32.16}{4^2 \times \frac{1}{40} \times 15.7^2} = 5,288$ lb. $= 2.644$ tons. Ans.

In the above solution the weight of the hub and arms has been neglected, and the whole weight has been assumed as being concentrated in the rim. This is the usual engineering practice. The rim is made heavy enough to make the engine run within the required limits of steadiness, and the weight of the arms, hub, etc., whatever it may be, simply adds so much more to the steadiness of running.

REVERSING GEAR.

1329. In many engines, it is necessary that the ratio of expansion be often varied, and that the direction of working be often changed. An enormous number of mechanisms have been devised to accomplish this result. The earliest, as well as one of the most efficient, is the **Stephenson link motion,** shown in Fig. 316.

1330. Two eccentrics A and B are keyed to the shaft. The two eccentric rods E and F are fastened to a link L. The valve stem V is bolted to a block W, which fits into L.

When the mechanism is in the position shown in the figure, the eccentric rod E is in line with the valve stem, and the action is precisely similar to the action of an engine with a single eccentric. The eccentric A will govern the steam distribution, and B will simply have no effect whatever. The angular advance is $A\,O\,C$; consequently, the engine will run in the direction shown by the arrow.

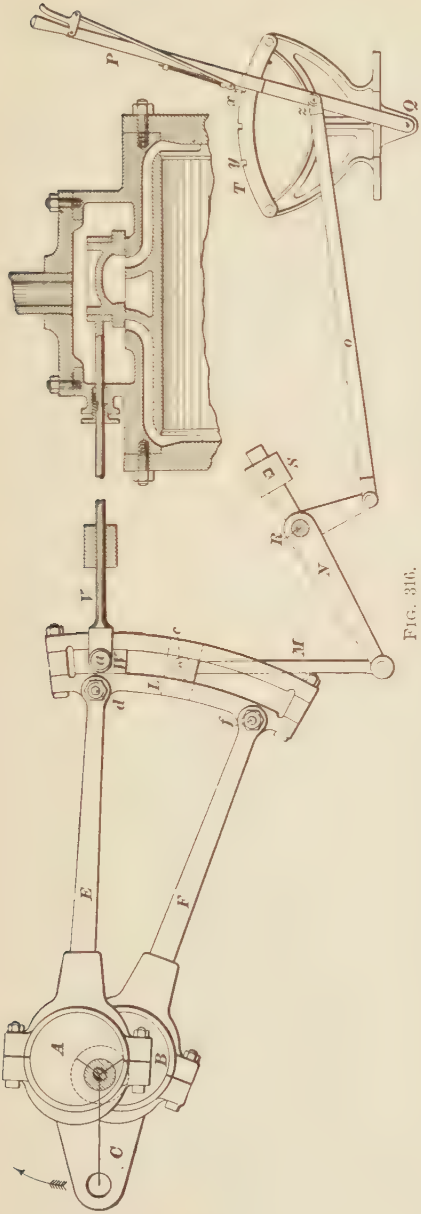


FIG. 316.

By means of the reversing lever *P*, and the links *O*, *N*, and *M*, pivoted at *R*, the link *L* can be raised until the eccentric rod *F* is in line with the valve stem *V*. The eccentric *B* will then impart motion to the valve; *B O C* will be the angular advance, and, since the eccentric must be ahead of the crank, the latter must move in the direction opposite to that shown by the arrow.

If the link be raised until the point of suspension *c* is just opposite the point *a* of the valve stem, the valve must partake equally of the motion of both eccentrics, and, hence, the engine will run in neither direction. If the link be raised so that *a* occupies a position somewhere between *c* and *d*, the engine will still run in the direction shown in the figure, but the motion of the point *a* will be influenced in some degree by the motion of *F*, and the travel of the valve will be less than when in the position of the figure. The result

is an earlier cut-off and compression. In locomotive engines, the sector T has several notches. When starting, the lever is thrown into the last notch, thus giving the valve the longest possible travel and the engine the greatest power. After the train is started and the power required is less, the lever is thrown nearer the center, thus decreasing the travel of the valve, and, in consequence, the power developed by the engine.

STRENGTH OF MATERIALS.

MATERIALS USED FOR CONSTRUCTION.

1331. The principal materials used in engineering construction are *timber*, *brick*, *stone*, *cast iron*, *wrought iron*, and *steel*. Table 23 gives their average weights per cubic foot.

TABLE 23.

Material.	Average Weight. Pounds per Cubic Foot.	Approximate Weight of Piece 1″ Square and 1′ Long in Lb.
Timber	40	.278
Brick	125	
Stone	160	
Cast Iron	450	3.125
Wrought Iron	480	3.333⅓
Steel	490	3.403

CAST IRON.

1332. **Cast iron** is a combination of pure iron with from 2% to 6% of carbon.

Pig iron is the result of the first smelting, and is obtained directly from the ore. Pig iron is rarely, if ever, used for anything except to be remelted and made into cast iron or wrought iron.

1333. Cast iron is of two kinds, *white cast iron* and *gray cast iron.* The first is a chemical compound of iron with from 2% to 6% of carbon, nearly all the carbon being chemically combined with the iron. The second, or gray cast iron, contains a part of the carbon in chemical combination, and the rest in the state of graphite mechanically mixed with the iron. When a piece of gray cast iron is

broken, a large number of black specks are seen on the broken ends; these specks are pure carbon in the form of graphite.

1334. White cast iron contains hardly any free carbon. It is of two kinds, granular and crystalline. Both are very hard and brittle, and are only used for conversion into wrought iron or steel.

1335. Gray cast iron is divided into three classes, known as Nos. 1, 2, and 3.

No. 1 contains the largest amount of carbon in mechanical mixture, which makes it soft and fusible, though not as strong as Nos. 2 and 3. It is very suitable for making castings where precision in form is required, as, owing to its fluidity, it fills the mold well; but it is not suitable for castings requiring strength.

No. 2 is most suitable for use in constructions, as it is stronger than No. 1, and not so soft.

No. 3 contains the smallest amount of carbon in the graphitic (uncombined) form, and is, in consequence, harder and more brittle. It is fit only for the massive and heavy parts of machinery.

1336. Cast iron has certain advantages and disadvantages as a material for engineering construction. It is easy to give it any desired form. A pattern of the piece desired is made; a mold is made in the sand, the pattern is removed, and the melted iron poured in. Cast iron resists oxidation (rust) better than either wrought iron or steel. Its compressive (crushing) strength is very high, but its tensile (stretching) strength is comparatively low. It cannot be riveted, or welded by forging. It is brittle, breaking off without giving much warning, and stretching but little before giving away. It is liable to have hidden and small surface defects and air bubbles, and this makes its strength uncertain.

Another serious drawback in the use of cast iron is its liability to initial stresses from inequality in cooling after having been poured into the molds. Thus, if one part of

the casting is very thin and another very thick, the thin part cools first, and, in cooling, contracts; the thick part, cooling afterwards, causes stresses in the thin part, which may be sufficient to break it, or, if not, there may be so great a stress in the thin part that a small additional force will break it.

Cast iron is not well adapted to a tensile stress, nor to resist shocks. It is used for columns and posts in buildings, on account of its high compressive strength. In machinery, it is used in all those parts where weight, mass, or form is of more importance than strength, as in frames and bed plates of machines, and for hangers, pulleys, gear wheels, etc. It is also used for water mains where the pressure to be resisted is not too great.

1337. Malleable cast iron is made by heating the casting in an annealing oven, in powdered hematite ore. It can be hammered into any desired shape when cold, but is very brittle when hot.

WROUGHT IRON.

1338. Wrought iron is the product resulting from the reduction of the carbon in cast iron. It is obtained by melting white cast iron, and passing an oxidizing flame over it. When the carbon is burned out, the mass of iron is left in a pasty condition, full of holes. It is then taken out, and hammered or rolled in order to unite it into one mass. The result of this first process is not suitable for use in any construction of importance, and has to be reheated and rerolled a number of times, in order to make it homogeneous, and to remove flaws from within the iron.

At best, therefore, wrought iron is a series of welds; if a piece is broken, the separate layers of which it is composed can be seen plainly. It cannot be melted and run into molds, as can be done with cast iron; but it can be easily welded by forging; that is, two pieces of wrought iron can be united by raising them to the proper temperature and hammering them together. Wrought iron is much more capable of bearing a tensile or transverse stress than cast

iron; it is tougher, stretches more, and gives more warning before fracture.

It withstands shocks far better than cast iron. Two pieces may be punched or drilled and riveted together. The entire process weakens the iron, and cast iron would not withstand it. It has also to withstand flanging and the stresses due to changes of temperature.

Wrought iron cannot be hardened like steel by heating and then dipping in water, but may be *case-hardened* by rubbing the surface with potassium cyanide or potassium ferrocyanide while at a cherry-red heat and then dipping in water. The cyanide causes the iron to be carbonized to a slight depth; that is, through a depth of about $\frac{1}{32}$ of an inch the iron is converted into steel which can be hardened. Cast iron may be hardened in the same way.

The quality of wrought iron varies considerably, and the terms by which it is known in the market refer to the amount of working which the iron has received. We thus have common bar iron, best iron, double best, and triple best. These terms are only rough indications of quality. When wrought iron is rolled cold under great pressure, it has a smooth polished surface, and its strength is greatly increased.

When the word iron is used alone, wrought iron is meant.

STEEL.

1339. Steel is a chemical compound of iron and carbon; that is, it contains no carbon in a free state, as cast iron does. Its tensile strength is greater than that of wrought iron, and its compressive strength greater than that of cast iron. It is by far the strongest material used in the mechanic arts. Its strength varies greatly with its purity and the amount of carbon it contains. According to the amount of carbon in it, steel is divided into high grade, medium grade, and low grade, the high grades having the most carbon. Steel, unlike wrought iron, is fusible; unlike cast iron, it can be forged; and, with the exception of the higher grades,

it can be welded by heating and hammering, although care must be exercised in so doing.

1340. The special characteristic of steel (the very lowest grades excepted) is that, when it is raised to a cherry red heat and suddenly cooled, it becomes brittle and exceedingly hard, and that by subsequent heating and slow cooling, the hardness may be reduced to any desired degree down to the point of least hardness that steel possessing that amount of carbon can have. The first process is called *hardening*, and the second *tempering*.

If the surface of a piece of steel that has been hardened is polished slightly so as to remove the dark scale or soot which covers it, and is then reheated, it will be found that, as the temperature increases, a series of colors appear. These colors are always the same for the same temperature, and, if the steel is suddenly cooled when one of the colors appears, it acquires a degree of hardness which is always the same for the same color and for the same quality of steel.

In Table 24 are given the colors with the corresponding temperatures that occur in tempering different kinds of tools.

TABLE 24.

Tools.	Color.	Temperature, Fahr.
Lancets	Pale yellow	430°
Razors and scalpels	Pale straw	450°
Penknives, chisels for cast iron, and screw taps	Yellow	470°
Scissors and chisels for wrought iron	Brown	490°
For carpenters' tools in general	Red	510°
Fine watch springs and table knives	Purple	530°
Swords and lock springs	Blue, bright	550°
Daggers, fine saws and needles	Blue, full	560°
Common saws and springs	Blue, dark	600°

Steel that has been hardened may be softened by heating it to a hardening temperature and then cooling it very slowly; this process is called **annealing.**

1341. Steel is made in one of the three following ways:

1. By adding carbon to wrought iron.
2. By removing carbon from cast iron.
3. By melting together cast and wrought iron in suitable proportions.

Several processes, varying with the quality of the product desired and the use for which it is intended, are used in making steel. The processes in general use are the following:

(*a*) **Cementation,** in which bars of very pure wrought iron are heated to a high temperature in contact with carbon. The product, known as **blister steel,** is used for cutlery, tools, etc.

(*b*) **Crucible steel,** also called **cast steel,** is made by melting pure wrought iron in a crucible with enough charcoal and cast iron to introduce the required amount of carbon. It is used for making springs, cutlery, tools, etc.

(*c*) **Bessemer steel** is made by decarbonizing cast iron by forcing a powerful blast of air through a melted mass of the iron. This removes the greater part of its carbon. A small quantity of very pure cast iron, rich in carbon, is then added, bringing up the percentage of carbon to the required amount.

(*d*) **Open-hearth steel** is made by fusing a charge consisting of the suitable proportions of cast iron with wrought iron scrap, or with Bessemer steel scrap.

1342. Bessemer and open-hearth steel contain more impurities than blister and crucible steel do; but they are much cheaper, and are just as suitable for many purposes. It is only in consequence of the introduction of these two cheap varieties that steel can be extensively used, as blister and crucible steels would, in the majority of cases, be too expensive.

Bessemer and open-hearth steels contain from .05% to $1\frac{1}{2}$% of carbon. The proportion of carbon in the best kinds of tool and cutlery steels is as follows:

Razor steel,	$1\frac{1}{2}$%.	Very difficult to forge, and easily burnt.
Saw file,	$1\frac{3}{8}$%.	Bears heat not above cherry redness.
Tool steel,	$1\frac{1}{4}$%.	Ordinary cutting tools. Welds with difficulty.
" "	$1\frac{1}{8}$%.	For mandrils and heavy cutting tools.
" "	1%.	For chisels, gravers, etc.

At the Imperial Works at Neuberg, Austria, the following percentages of carbon are present in the different grades of Bessemer steel:

1.58% to 1.38%.	Cannot be welded, and is rarely used.
1.38% " 1.12%.	Great care must be used in working.
1.12% " .88%.	Welds easily; used for bits, chisels, etc.
.88% " .62%.	Used for cutting tools, files, etc.
.62% " .38%.	Mild steel; for tires, etc.
.38% " .15%.	Tempers slightly; for boiler plates and axles.
.15% " .05%.	Does not temper; steel for pieces of machinery.

1343. It will be noticed from what precedes that the hardness of steel depends upon the amount of carbon it contains.

Some kinds of crucible cast steel can be hardened by heating to a low red heat and then allowing them to cool slowly in the air without dipping in water. They are called self-hardening steels, the best known being Mushet's special tool steel. This contains about 2% carbon with 7% to 12% tungsten in alloy with the iron. The same property is characteristic of Hadfield's manganese steel, which contains between .8% and 1.2% of carbon and 7% to 20% of manganese.

As a rule, when a piece of steel is broken across the grain, the finer the grain and the whiter and cleaner the fracture the more carbon it contains.

STRESSES AND STRAINS.

1344. The molecules of a solid or rigid body being held together by the force of cohesion, this force must be overcome to a greater or less degree in order to change the form and size of the body, or to break it into parts. The internal resistance which a body offers to any force tending to overcome the force of cohesion is called a **stress.** If a weight of 1,000 pounds is held in suspension by a rod, there will be a stress of 1,000 pounds in the rod. In this country and England, stresses are measured in pounds or tons; in nearly all other civilized countries, in kilograms. Whenever a body is subjected to a stress, the total stress induced by the acting force at any section of the body is the same as the total stress at any other section.

1345. The **unit stress** (called also the **intensity** of stress) is the stress per unit of area; or, it is the total stress divided by the area of the cross-section. In the above illustration, if the area had been 4 sq. in. the unit stress would have been $\frac{1,000}{4} = 250$ lb. per sq. in. Had the area been $\frac{1}{2}$ sq. in., the unit stress would have been $\frac{1,000}{.5} = 2,000$ lb. per sq. in.

Let $P =$ the total stress in pounds;
$A =$ area of cross-section in square inches;
$S =$ unit stress in pounds per square inch.

Then, $$S = \frac{P}{A}, \text{ or } P = A S. \qquad \textbf{(108.)}$$

That is, *the total stress equals the area of the section, multiplied by the unit stress.*

When a body is stretched, shortened, or in any way deformed through the action of a force, the amount of deformation is called a **strain.** Thus, if the rod before mentioned had been elongated $\frac{1}{10}''$ by the load of 1,000 pounds, the strain would have been $\frac{1}{10}''$. Within certain limits, to be given hereafter, strains are proportional to the stresses producing them.

1346. The **unit strain** is the strain per unit of length or of area, but is usually taken per unit of length and called the *elongation* per unit of length. In this paper, the unit of length will be considered as one inch. The unit strain, then, equals the total strain divided by the length of the body in inches.

Let l = length of body in inches;
e = elongation in inches;
s = unit strain.

Then, $$s = \frac{e}{l}, \text{ or } e = ls. \qquad \textbf{(109.)}$$

1347. Whenever a force, no matter how small, acts upon a body, it produces a stress and a corresponding strain.

According to the manner in which forces act upon a body, the stresses are divided into the following classes:

1. *Tension*, which produces a tensile or pulling stress.
2. *Compression*, which produces a compressive or crushing stress.
3. *Shear*, which produces a shearing or cutting stress.
4. *Torsion*, which produces a torsional or twisting stress.
5. *Flexure*, which produces a transverse or bending stress.

TENSION.

1348. When two forces act upon a body in opposite directions (away from each other) the body is said to be in tension. The two forces tend to elongate the body and thus produce a tensile stress and strain. A weight supported by a rope affords a good example. The weight acts downwards, and the reaction of the support to which the upper end of the rope is fastened acts upwards; the result being that the rope is stretched more or less, and a tensile stress is produced in it. Another familiar example is the connecting rod of a steam engine on the return stroke. The cross-head then exerts a pull on one end of the rod, which is resisted by the crank-pin on the other.

EXAMPLE.—An iron rod, 2 inches in diameter, sustains a load of 90,000 pounds; what is the unit stress?

SOLUTION.—Using formula **108**,

$$P = A\,S, \text{ or } S = \frac{P}{A} = \frac{90,000}{2^2 \times .7854} = 28,647.82 \text{ lb. per sq. in. Ans.}$$

EXPERIMENTAL LAWS.

1349. The following laws have been established by experiment:

1. *When a body is subjected to a small stress, a small strain is produced, and when the stress is removed the body springs back to its original shape. This leads to the conclusion that, for small stresses, bodies are perfectly elastic.*

2. *Within certain limits, the change of shape (strain) is directly proportional to the applied force.*

3. *When the stress is sufficiently great, a strain is produced which is partly permanent; that is, the body does not spring back entirely to its original form when the stress is removed. This lasting part of the strain is called a* **set,** *and in such cases the strain is not proportional to the stress.*

4. *Under a still greater stress, the strain rapidly increases, and the body is finally ruptured or broken.*

5. *A force acting suddenly, as a shock, causes greater injury than a force gradually applied.*

According to the first law, the body will resume its original form when the force is removed, provided the stress is not too great. This property is called *elasticity.* According to the second law, the strain is proportional to the stress within certain limits. Thus, if a pull of 1,000 pounds elongates a body .1″, a pull of 2,000 pounds would elongate it .2″. This is true up to a certain limit, beyond which the body will not resume its original form upon the removal of the stress, but will be permanently strained more or less, according to the amount of stress. The stress at the point where the set begins is called the **elastic limit.** All strains produced by stresses within (less than) the elastic limit are directly proportional to the stresses.

1350. Stresses are equal but opposed to the external forces producing them, and are, therefore, measured and represented by these forces. Thus, as we have explained before, a force of 1,000 pounds produces a stress of 1,000 pounds. The external force is the force applied to a fixed body; the stress is the resistance offered by the body to a change of form; and when the body ceases to change (as when a rod ceases to elongate), the stress just balances the external force.

COEFFICIENT OF ELASTICITY.

1351. Amongst engineers, the term *elasticity* means the *resistance* which a body offers to a permanent change of form; and by *strength*, the resistance which a body offers to division or separation into parts.

It follows from this that those bodies which have the highest elastic limit are the most elastic.

1352. The **coefficient of elasticity** is the ratio of the unit stress to the unit strain, provided the elastic limit is not exceeded. Let S be the unit stress, s the unit strain, and E the coefficient of elasticity; then, by definition, $E = \frac{S}{s}$. Substituting the values of S and s obtained from formulas **108** and **109**,

$$E = \frac{S}{s} = \frac{P}{A} \div \frac{e}{l} = \frac{P l}{A e}. \qquad \textbf{(110.)}$$

1353. If in this formula we assume $e = l$, and $A = 1$ (1 square inch), then $E = P$. That is, the coefficient of elasticity is that force which, *if stress and strain continued proportional to each other*, would produce in a bar of unit area a strain equal to the original length of the bar ($l = e$). This, however, is never the case, as the elastic limit and the ultimate strength are reached before the applied force reaches the value E.

EXAMPLE.—A wrought iron bar 2 inches square and 10 feet long is stretched .0528 inch by a stress of 44,000 pounds; what is the coefficient of elasticity?

SOLUTION.—Using formula **110**,

$$E = \frac{P l}{A e} = \frac{44{,}000 \times 10 \times 12}{2^2 \times .0528} = 25{,}000{,}000 \text{ lb. per sq. in.} \quad \text{Ans.}$$

1354. The **ultimate strength** of any material is that unit stress which is just sufficient to break it.

1355. The **ultimate elongation** is the total elongation produced in a unit of length of the material having a unit of area, by a stress equal to the ultimate strength of the material.

1356. For the same size, quality, and kind of material, the ultimate strength, ultimate elongation, coefficient of elasticity, and elastic limit are the same for different pieces. Table 25 gives the *average* values of the coefficient of elasticity (E_1), elastic limit (L_1), ultimate strength (S_1), and

TABLE 25.

Material.	Coefficient of Elasticity. E_1.	Elastic Limit. L_1.	Ultimate Tensile Strength. S_1.	Ultimate Elongation. s_1.
	Lb. per Sq. In.	Lb. per Sq. In.	Lb. per Sq. In.	In. per Linear Inch.
Timber......	1,500,000	3,000	10,000	0.015
Cast Iron....	15,000,000	6,000	20,000	0.005
Wrought Iron	25,000,000	25,000	55,000	0.20
Steel........	30,000,000	50,000	100,000	0.10

ultimate elongation (s_1), of different materials, the quantities given being for *tension* only. As brick and stone are never used in tension, their values are not given.

The values in this table are subject to great variation, and cannot be depended upon in designing machine parts. Thus, the ultimate tensile strength of steel varies from less than 60,000 to more than 180,000 pounds per square inch, according to its purity and the amount of carbon it contains; that of cast iron from 12,000, or 13,000, to over 40,000; wrought iron varies from 40,000 to 72,000, according to quality, the latter value being for iron wire. Timber varies fully as much as, if not more than, any of the three preceding materials, its properties depending upon the kind of wood, its degree of dryness, the manner of drying, etc.

All the problems in this section will be solved by using the average values given in the preceding and following tables; the designer, however, should not use them, but either test the materials himself or state in the specifications what strength the material must have. For example, mild steel, for boiler shells, should have a tensile strength of not less than 60,000 or 65,000 pounds per square inch; Bessemer steel, for steel rails, not less than 110,000; open-hearth steel, for locomotive tires, not less than 125,000, and crucible cast steel, for tools, cutlery, etc., not less than 150,000. It is also customary to specify the amount of elongation. This is necessary because, as a rule, the elongation decreases as the tensile strength increases. Having tested the material about to be used, or having specified the lowest limits, the designer can ascertain the strength and stiffness of construction by means of the formulas and rules which are to follow.

EXAMPLE.—How much will a piece of steel 1 inch in diameter and 1 foot long elongate under a steady load of 15,000 pounds?

SOLUTION.—$E_1 = \frac{P\,l}{A\,e}$, or $e = \frac{P\,l}{A\,E_1}$.

From Table 24, $E_1 = 30{,}000{,}000$ for steel; hence,

$$e = \frac{15{,}000 \times 12}{1^2 \times .7854 \times 30{,}000{,}000} = .00764''. \quad \text{Ans.}$$

NOTE.—All lengths given in this treatise on Strength of Materials must be reduced to inches before substituting in the formulas.

EXAMPLE.—A piece of timber has a cross-section $2'' \times 4''$ and is 6 feet long. A certain stress produces an elongation of .144 inch; what is the value of the stress in pounds?

SOLUTION.—

$$E_1 = \frac{P\,l}{A\,e}, \text{ or } P = \frac{E_1\,A\,e}{l} = \frac{1{,}500{,}000 \times 2 \times 4 \times .144}{6 \times 12} = 24{,}000 \text{ lb.} \quad \text{Ans.}$$

COMPRESSION.

1357. If the length of the piece is not more than five times its least transverse dimension (its diameter, when round; its shorter side, when rectangular, etc.), the laws of compression are similar to those of tension. The strain is proportional to the stress until the elastic limit has been

reached; after that, it increases more rapidly than the stress, as in the case of tension. The area of the cross-section is slightly enlarged under compression. In Table 26 are given the average compression values of E, L, and S for wood, brick, stone, cast iron, wrought iron, and steel. (See also Table 25.) E is not given for brick, nor L for cast iron, brick, or stone, because these values are not known. To distinguish between tension and compression when applying a formula, E_2, L_2, and S_2 will be used instead of E_1, L_1, and S_1.

TABLE 26.

Material.	Coefficient of Elasticity. E_2.	Elastic Limit. L_2.	Ultimate Compressive Strength. S_2.
	Lb. per Sq. In.	Lb. per Sq. In.	Lb. per Sq. In.
Timber..........	1,500,000	3,000	8,000
Brick	....	......	2,500
Stone............	6,000,000	......	6,000
Cast Iron.........	15,000,000	......	90,000
Wrought Iron.....	25,000,000	25,000	55,000
Steel.............	30,000,000	50,000	150,000

1358. When the length of a piece subjected to compression is greater than ten times its least transverse dimension, it is called a *column*, and the material fails by a sidewise bending or flexure. The preceding table is to be used only for pieces whose length does not exceed *five* times the least dimension of the cross-section.

EXAMPLE.—How much will a wrought iron bar 4 inches square and 15 inches long shorten under a load of 100,000 pounds?

SOLUTION.—$E_2 = \frac{P\,l}{A\,e}$, or $e = \frac{P\,l}{A\,E_2}$.

Hence, $e = \frac{100,000 \times 15}{16 \times 25,000,000} = .00375''$. Ans.

SHEAR.

1359. When two surfaces move in opposite directions very near together in such a manner as to cut a piece of material, or to pull part of a piece through the remainder, the piece is said to be *sheared*. A good example of a shearing stress is a punch; the two surfaces in this case are the bottom of the punch and the top of the die. Another example is a bolt with a thin head; if the pull on the bolt is great enough, it will be pulled through the head and leave a hole in it, instead of the bolt breaking by pulling apart, as would be the case with a thick head. In this case, the two surfaces are the under side of the head and the surface pressed against. Other examples are a knife cutting a piece of wood, and the ordinary shears from which this kind of stress takes its name.

TABLE 27.

Material.	Coefficient of Elasticity. E_3.	Ultimate Shearing Strength. S_3.
Timber (across the grain).....	....	3,000
Timber (with the grain)	400,000	600
Cast Iron....................	6,000,000	20,000
Wrought Iron................	15,000,000	50,000
Steel	....	70,000

1360. Formula **108** applies in cases of shearing stress, but formulas **109** and **110** are never used for shearing. In the preceding table, E_3 and S_3 are used to represent, respectively, the coefficient of shearing elasticity, and ultimate shearing strength.

EXAMPLE.—What force is necessary to punch a one-inch hole in a wrought iron plate $\frac{3}{8}$ of an inch thick?

SOLUTION.—$1'' \times 3.1416 \times \frac{3}{8}'' = 1.1781$ sq. in. = area of punched surface = area of a cylinder $1''$ in diameter and $\frac{3}{8}''$ high. Using formula **108**,

$$P = A\,S_3 = 1.1781 \times 50{,}000 = 58{,}905 \text{ lb.} \quad \text{Ans.}$$

EXAMPLE.—A wooden rod 4 inches in diameter and 2 feet long is turned down to 2 inches diameter in the middle so as to leave the enlarged ends each 6 inches long. Will a steady stress pull the rod apart in the middle, or shear the ends?

SOLUTION.—$P = A\,S_3 = 2 \times 3.1416 \times 6 \times 600 = 22{,}620$ lb. to shear off the ends.

The force required to rupture by tension is

$$P = A\,S_1 = 2^2 \times .7854 \times 10{,}000 = 31{,}416 \text{ lb.}$$

Since the former is only about $\frac{3}{4}$ of the latter, the piece will fail through the shearing off of the end. Ans.

Had a transverse stress been used, the force necessary to shear off a section of the end would have been

$$4^2 \times .7854 \times 3{,}000 = 37{,}700 \text{ lb.}$$

FACTORS OF SAFETY.

1361. It was previously stated that no stress should ever be applied to a machine part that would strain it beyond the elastic limit. The usual practice is to divide the ultimate strength of the material by some number depending upon the kind and quality of the material, and upon the nature of the stress; this number is called a **factor of safety.**

The factor of safety for any material is the ratio of its ultimate strength to the actual stress to which it is subjected, or for which it is intended.

In Table 27, 70,000 pounds per square inch is given as the ultimate shearing strength for steel. Now, suppose that the actual stress on a piece of steel is 10,000 pounds per square inch; then, the factor of safety for this piece would be $\frac{70{,}000}{10{,}000} = 7$.

1362. To find the proper allowable working strength of a material, divide the ultimate strength for tension, compression, or shearing, as the case may be, by the proper factor of safety.

Table 28 gives the factors of safety generally used in American practice. Factors of safety will always be denoted by the letter f in the formulas to follow.

TABLE 28.

Material.	For Steady Stress. (Buildings.)	For Varying Stress. (Bridges.)	For Shocks. (Machines.)
Timber...........	8	10	15
Brick and Stone...	15	25	30
Cast Iron.........	6	10	15
Wrought Iron.....	4	6	10
Steel.............	5	7	10

1363. *Twice as much strain is caused by a suddenly applied stress as by one that is gradually applied.* For this reason a larger safety factor is used for shocks than for steady stresses. In general, the factor of safety for a given material must be chosen according to the nature of the stress.

The designer usually chooses his own factors of safety. If the material has been tested, or the specifications call for a certain strength, then the factor of safety can be chosen accordingly.

Example.—Assuming the mortar and brick to be of the same strength, how many tons could be safely laid upon a brick column 2 feet square and 8 feet high?

Solution.—$P = A\ S_2 = 2 \times 2 \times 144 \times 2{,}500 = 1{,}440{,}000$ lb. $= 720$ tons. The factor of safety for this case is 15 (see Art. **1362** and Table **28**); hence, $720 \div 15 = 48$ tons. Ans.

Example.—What must be the diameter of the journals of a wrought iron locomotive axle to resist shearing safely, the weight on the axle being 40,000 pounds?

Solution.—Let f be the factor of safety; then, $P = \frac{A\ S_3}{f}$, or $A = \frac{Pf}{S_3}$. Since the axle has two journals, the stress on each journal is 20,000 lb. Owing to inequalities in the track, the load is not a steady one, but varies; for this reason, the factor of safety will be taken as 6. Then, $A = \frac{20{,}000 \times 6}{50{,}000} = 2.4$ sq. in. Therefore, $d = \sqrt{\frac{2.4}{.7854}} = 1\frac{3}{4}''$. Ans.

EXAMPLE.—Considering the piston rod of a steam engine as if its length were less than ten times its diameter, what must be the diameter of a steel rod, if the piston is 18 inches in diameter and the steam pressure is 110 pounds per square inch ?

SOLUTION.—Area of piston is $18^2 \times .7854 = 254.47$ sq. in. $254.47 \times 110 =$ 27,991.7 lb., or, say, 28,000 lb. = stress in the rod. $A = \frac{Pf}{S_2} = \frac{28,000 \times 10}{150,000}$ $= 1.87''$, nearly, or, say, $1\frac{7}{8}''$. Ans.

1364. When designing a machine, care should be taken (1) *to make every part strong enough to resist any stress likely to be applied to it; and* (2) *to make all parts of equal strength.* The reason for the first statement is obvious, and the second should be equally clear, since no machine can be stronger than its weakest part (proportioned, of course, for the stress it is to bear), and those parts of the machine which are stronger than others contain an excess of material which is wasted. In actual practice, however, this second rule is frequently modified. Some machines are intended to be massive and rigid, and need an excess of material to make them so; in others, there are difficulties in casting that modify the rule, etc., etc. In most cases, the designer must rely on his own judgment.

EXAMPLES FOR PRACTICE.

1. A cast iron bar is subjected to a steady tensile stress of 120,000 pounds. The cross-section is an ellipse whose axes are 6 and 4 inches. (*a*) What is the stress per square inch? (*b*) What load will the bar carry with safety?

Ans. $\begin{cases} (a)\ 6,367.65 \text{ lb. per sq. in.} \\ (b)\ 62,832 \text{ lb.} \end{cases}$

2. How much will a piece of steel 2 inches square and 10 inches long shorten under a load of 300,000 pounds? Ans. .025″.

3. A cylindrical wooden pin $1\frac{3}{4}$ inches in diameter is subjected to a double shearing stress. If the stress is suddenly applied, what total force is necessary to shear the pin? Ans. 14,431 lb.

4. A wrought iron tie rod is $\frac{3}{4}$ inch diameter; how long must it be to lengthen $\frac{3}{8}$ inch under a steady pull of 5,000 pounds? Ans. 69 ft.

5. A steel bar having a cross-section of $5'' \times 4''$ and 14 feet long is lengthened .036 inch by a steady pull of 120,000 pounds; what is its coefficient of elasticity? Ans. 28,000,000 lb. per sq. in.

6. Which is the stronger, weight for weight, a bar of chestnut wood whose tensile strength is 12,000 pounds per square inch and specific gravity .61, or a bar of steel whose tensile strength is 125,000 pounds per square inch?

7. What should be the diameter of a cast-iron pin subjected to a suddenly applied double shearing stress of 40,000 pounds to withstand the shocks with safety? Ans. $4\frac{5}{8}''$, nearly.

8. What safe steady load may be placed upon a brick column 2 feet square and 9 feet high? Ans. 96,000 lb.

PIPES AND CYLINDERS.

1365. A pipe or cylinder subjected to a pressure of steam or water is strained equally in all its parts, and, when rupture occurs, it is in the direction of its length.

Let $d =$ inside diameter of pipe in inches;
$l =$ length of pipe in inches;
$p =$ pressure in pounds per square inch;
$P =$ total pressure.

Then, $P = p\,l\,d$.

This formula is derived from a principle of hydrostatics that the pressure of water in any direction is equal to the pressure on a plane perpendicular to that direction. In Fig. 317, suppose the direction of pressure to be as shown by the arrows; $A\ B$ would then be the plane perpendicular to this direction, the width of the plane being equal to the diameter, and the length equal to the length of the pipe. The area of the plane would then be $l \times d$, and the total pressure $P = p \times l \times d$, as above.

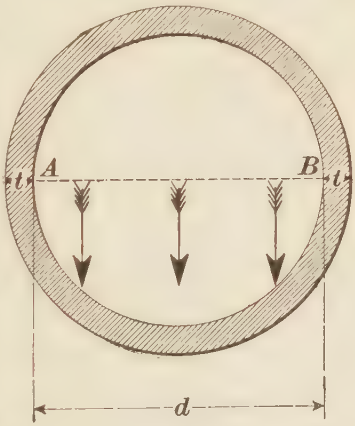


FIG. 317.

Suppose the pipe to have a thickness t, and let S be the working strength of the material; then, the resistance of the pipe on *each* side is $t\,l\,S$. Resistance must equal pressure; therefore, $p\,l\,d = 2\,t\,l\,S$, or

$$p\,d = 2\,t\,S. \qquad \textbf{(111.)}$$

Also, $p \frac{2tS}{d}$, which is the maximum pressure a pipe of a given material and of given dimensions can stand.

The pressure of water per square inch may be found by the formula, $p = .434\ h$, where h is the head in feet. In pipes where shocks are likely to occur, the factor of safety should be high. The thickness of a pipe to resist a given pressure varies directly as its diameter, the pressure remaining constant.

EXAMPLE.—Find the factor of safety for a cast-iron water pipe 12 inches in diameter and $\frac{3}{4}$ inch thick, under a head of 350 feet.

SOLUTION.—Here p, pressure per square inch, equals $.434\ h = .434 \times 350 = 151.9$ lb. Substituting, in formula **111,** the values given,

$$151.9 \times 12 = 2 \times \tfrac{3}{4} \times S, \text{ or } S = 1{,}215.2 \text{ lb. per sq. in.}$$

In Table 25, Art. **1356,** the ultimate tensile strength of cast iron is given as 20,000 pounds per square inch; then, the factor of safety is $f = \frac{20{,}000}{1{,}215.2} = 16+$. Ans. The pipe would, therefore, be secure against shocks.

EXAMPLE.—Find the proper thickness for a wrought-iron steam pipe 18 inches in diameter to resist a pressure of 140 pounds per square inch.

SOLUTION.—Using a factor of safety of 10, the working strength $S = \frac{55{,}000}{10} = 5{,}500$ lb. per sq. in. From formula **111,** $t = \frac{p\,d}{2S} = \frac{140 \times 18}{2 \times 5{,}500} = 0.23$ in. In practice, however, the thickness is made somewhat greater than the formula requires.

CYLINDERS.

1366. The tendency of a cylinder subjected to internal pressure is to fail or rupture in the direction of its length, the same as a pipe.

Let π (pronounced *pi*) be the ratio of the circumference of a circle to its diameter $= 3.1416$; then, $\frac{1}{4}\pi = .7854 =$ ratio of area to the square of the diameter.

The total pressure on the cylinder head $= \frac{1}{4}\pi d^2 p$. Let $S =$ working unit stress $= \frac{S_1}{f}$, then $\pi\, d\, t\, S =$ the resistance to rupture caused by the pressure acting on the opposite

cylinder heads and tending to elongate the cylinder. Since the resistance must equal the force, or pressure, $\frac{1}{4}\pi\, d^2\, p = \pi\, d\, t\, S$, or

$$p\, d = 4\, t\, S. \qquad \textbf{(112.)}$$

$$\text{Also, } p = \frac{4\, t\, S}{d}.$$

Since, for longitudinal rupture, $p = \dfrac{2\, t\, S}{d}$, it is seen that a cylinder is twice as strong against transverse rupture as against longitudinal rupture. Hence, other things being equal, the cylinder will always fail by longitudinal rupture.

1367. The foregoing formulas are for comparatively thin pipes and cylinders, in which the thickness is less than about $\frac{1}{10}$ inside radius. For pipes and cylinders whose thickness is greater than $\frac{1}{10}$ radius, use the following formula, in which r = the inner radius, and the other letters have the same meaning as before.

$$p = \frac{S\, t}{r + t}. \qquad \textbf{(113.)}$$

Substituting the values given in the example in Art. **1365,** in formula **113,** instead of formula **111,**

$$p = \frac{S\, t}{r + t}, \text{ or } S = \frac{p\,(r + t)}{t} = \frac{151.9 \times (6 + \frac{3}{4})}{\frac{3}{4}} = 151.9 \times 6\tfrac{3}{4} \times \tfrac{4}{3} = 1{,}367.1 \text{ lb.}$$

When formula **111** was used, $S = 1{,}215.2$ lb.; hence, formula **113** gives, for this case, a value $12\frac{1}{2}\%$, or $\frac{1}{8}$ greater.

The formula for spheres is the same as that for transverse rupture of cylinders, or $p\, d = 4\, t\, S$.

1368. A cylinder under external pressure is theoretically in a similar condition to one under internal pressure, so long as its cross-section remains a true circle. A uniform internal pressure tends to preserve the true circular form, but an external pressure tends to increase the slightest variation from the circle, and to render the cross-section elliptical. The distortion, when once begun, increases rapidly, and

failure occurs by the collapsing of the tube rather than by the crushing of the material. The flues of a steam boiler are the most common instances of cylinders subjected to external pressure.

The letters having the same meaning as before, the following formula gives the collapsing pressure in pounds per square inch for wrought-iron pipe:

$$p = 9,600,000 \frac{t^{2.18}}{l\,d}. \qquad \textbf{(114.)}$$

EXAMPLE.—What must be the thickness of a boiler tube 2 inches in diameter and 11 feet long, if the steam pressure is to be not over 160 pounds per square inch?

SOLUTION.—Using formula **114,** with a factor of safety of 10, and solving for t,

$$t = \sqrt[2.18]{\frac{10\,p\,l\,d}{9,600,000}} = \sqrt[2.18]{\frac{10 \times 160 \times 11 \times 12 \times 2}{9,600,000}} = \sqrt[2.18]{\frac{11}{250}}.$$

Hence, $\log t = \dfrac{\log 11 - \log 250}{2.18} = \bar{1}.37771$, or $t = .2386''$, say $\frac{1}{4}''$.

EXAMPLES FOR PRACTICE.

1. What must be the thickness of a 16-inch cast-iron stand pipe which is subjected to a head of water of 250 feet? Assume that the stress is steady. Ans. .26″.

2. What should be the thickness of a wrought-iron boiler flue 15 feet long, 4 inches in diameter, and subjected to an external pressure of 200 pounds per square inch? Ans. .42″.

3. What pressure per square inch can be safely sustained by a cast-iron cylinder 12 inches in diameter and 3 inches thick? Ans. 1,111$\frac{1}{9}$ lb. per sq. in.

4. What external pressure per square inch can a wrought iron pipe 20 feet long, 3 inches in diameter, and $\frac{3}{8}$ inch thick, safely sustain and be secure against shocks? Ans. 157.2 lb. per sq. in.

5. A cast-iron cylinder 14 inches in diameter sustains a total pressure of 125 tons; what is the necessary thickness, assuming that the pressure is gradually applied, and that the cylinder is not subjected to shocks? Ans. 6.65″.

6. A cylindrical boiler shell 3 feet in diameter is subjected to a steady hydrostatic pressure of 180 pounds per square inch. What should its thickness be if made of steel having a tensile strength of 60,000 pounds per square inch? Ans. .27″.

ELEMENTARY GRAPHICAL STATICS.

Before taking up the subject of flexure, some fundamental principles of Graphical Statics not heretofore considered will be explained and applied to the case of beams.

FORCE DIAGRAM AND EQUILIBRIUM POLYGON.

1369. In Arts. **878** and **879**, the polygon of forces was used to find the resultant of several forces having a common point of application, or whose lines of action passed through a common point. A method of finding the resultant will now be given when the forces lie, or may be considered as lying, in the same plane, but their lines of action do not pass through a common point.

In Fig. 318, let F_1, F_2, and F_3 be three forces whose magnitudes are represented by the lengths of their respective lines, and their directions by the positions of the lines and by

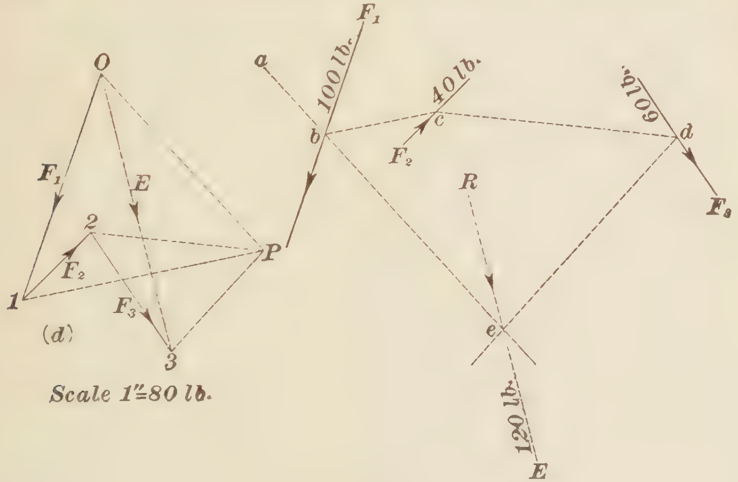


FIG. 318.

the arrow-heads. Construct the polygon of forces *O 1 2 3 O* as shown at (*d*), in the same manner as described in Art. **878**, *O 3* representing the direction and magnitude of the resultant. Everything is now known except the line on

which the point of application of the resultant must lie. To find this, proceed as follows:

Choose *any* point P, and draw $P\ O$, $P\ 1$, $P\ 2$, and $P\ 3$. Choose any point b, on the line of direction of one of the forces as F_1, and draw lines through b parallel to $P\ O$ and $P\ 1$, the latter intersecting F_2, or F_2 prolonged, in c. Draw $c\ d$ parallel to $P\ 2$, and intersecting F_3, or F_3 prolonged, in d. Draw $d\ e$ parallel to $P\ 3$, intersecting the line $a\ b\ e$, parallel to $P\ O$, in e. The point e is a point on the line of direction of the resultant of the three forces. Hence, through e, draw R parallel and equal to $O\ 3$ and acting in the same direction; it will be the resultant.

The method just described is applicable to any number of forces considered as acting in the same plane. The resultant can also be found when the forces act in different planes, but the method of finding it will not be described here.

The point P is called the **pole**; the lines $P\ O$, $P\ 1$, $P\ 2$, $P\ 3$ joining the pole with the vertexes of the force polygon are called the **strings** or **rays**; the **force diagram** is the figure composed of the force polygon, $O\ 1\ 2\ 3\ O$, the pole, and the strings. The polygon $b\ e\ d\ c\ b$ is called the **equilibrium polygon.**

Since the pole P may be taken anywhere, any number of force diagrams and equilibrium polygons may be drawn, all of which will give the same value for the resultant, and whose lines $d\ e$ and $a\ e$ will intersect on the resultant R. To test the accuracy of the work, take a new pole and proceed as before. If the work has been done correctly, $d\ e$ and $a\ e$ will intersect on R.

The equilibrium polygon gives an easy method of resolving a force into two components.

EXAMPLE.—In Fig. 319, let $F = 16$ pounds be the force, and let it be required to resolve it into two *parallel* components, A and B, at distances respectively of 5 feet and 15 feet from F. What will be the magnitudes of A and B?

SOLUTION.—Draw $O\ 1$ to represent $F = 16$ lb. Choose any convenient pole P, and draw the rays $P\ O$ and $P\ 1$. Take any point a on F.

and draw $a\,b$ parallel to $P\,O$, intersecting A in b, and $a\,c$ parallel to $P\,1$, intersecting B in c. Join b and c by the line $b\,c$. Through the pole P, draw $P\,d$ parallel to $b\,c$, intersecting $O\,1$ in d. Then $O\,d$ is the magnitude of A, measure, to the scale to which $O\,1$ was drawn, and $d\,1$ is the magnitude of B to the same scale.

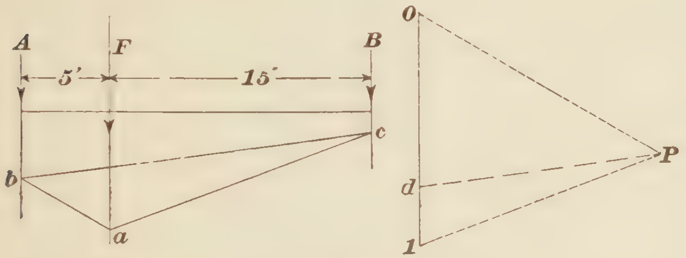


FIG. 319.

1370. If the components are not parallel to the given force, they must intersect its line of direction in a common point.

In Fig. 320, let $F = 16$ pounds be the force; it is required to resolve it into two components A and B, intersecting at a, as shown. Draw $O\ 1$ to some convenient scale equal to 16 pounds; then draw $O\,P$ and $1\,P$ parallel to A and B, and $O\,P$ and $P\,1$ are the values of the components A and B, respectively, both in magnitude and direction.

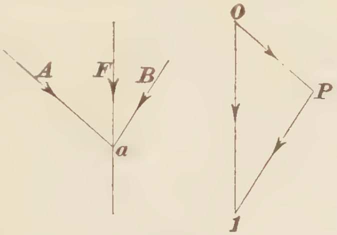


FIG. 320.

EXAMPLE.—Let F_1, F_2, F_3, F_4, and F_5, Fig. 321, be five forces whose magnitudes are 7, 10, 5, 12, and 15 pounds, respectively. It is required to find their resultant and to resolve this resultant into two components parallel to it and passing through the points a and b.

SOLUTION.—Choose any point O, Fig. 321, and draw $O\,1$ parallel and equal to F_1; 1–2 parallel and equal to F_2, etc.; $O\,5$ will be the value of the resultant, and its direction will be from O to 5, opposed to the other forces acting around the polygon. Choose a pole P, and complete the force diagram. Choose a point c on F_1, and draw the equilibrium polygon $c\,d\,e\,f\,g\,h\,c$; the intersection of $c\,h$, parallel to $P\,O$, and $g\,h$, parallel to $P\,5$, gives a point h, on the resultant R. Through h, draw R parallel to $O\,5$, and it will be the position of the line of action

of the resultant of the five forces. The components must pass through the points a and b, according to the conditions; hence, through a and b, draw V_1 and V_2 parallel to R. Since $O\,5$ represents the magnitude of R, draw $h\,k$ and $h\,l$ parallel to $P\,O$ and $P\,5$, respectively, as in Fig.

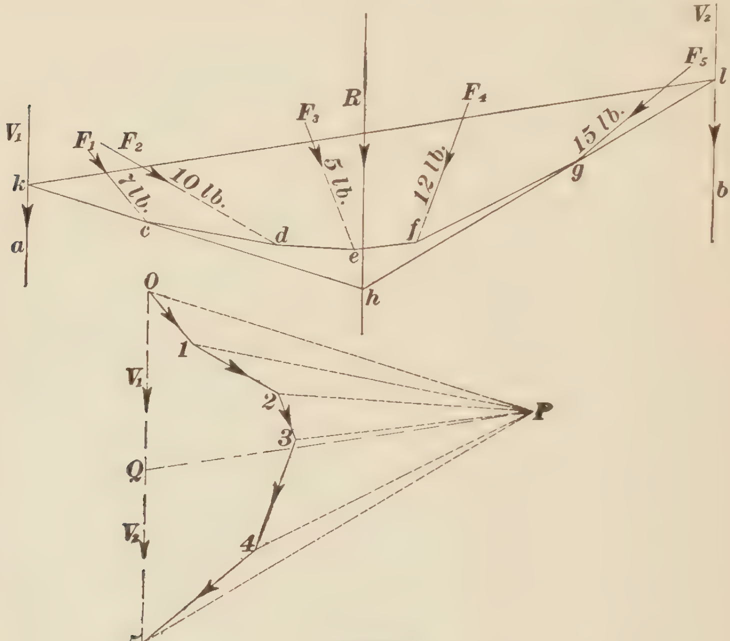


FIG. 321.

319 (they, of course, coincide with $c\,h$ and $g\,h$, since the same pole P is used), intersecting V_1 and V_2 in k and l. Join k and l, and draw $P\,Q$ parallel to $k\,l$. Then, $O\,Q = V_1$, and $Q\,5 = V_2$.

COMPOSITION OF MOMENTS.

1371. In Art. **906** it was stated that the moment of a force about a point is the product of the magnitude of the force by the perpendicular distance from the point to the line of action of the force. A force can act in two ways upon a body: it can either produce a motion of translation—that is, cause all the points of the body to move in straight parallel lines—or it can produce a motion of rotation—that is, make the body turn. A *moment* measures the capacity of a force to produce rotation about a given point. For example,

suppose, in Fig. 322, that $A\ C$ is a lever 30 inches long, having a fulcrum at B 10 inches from A. If a weight is suspended from C, it will cause the bar to rotate about B in the direction of the arrow. A weight suspended from A will cause it to revolve in the opposite direction, as indicated by the arrow. Suppose, for simplicity, that the bar itself weighs nothing. If two weights of 12 pounds each are hung at A and C, it is evident that the bar will revolve in the direction of the arrow at C, on account of the arm $B\ C$ being longer than the arm $A\ B$. Let the weight at A be increased until it equals 24 pounds. The bar will then balance exactly, and any additional weight at A will cause the bar to rotate in the opposite direction, as shown by the arrow at that point. When the lever is balanced, it will be found that $24 \times 10 = 12 \times 20$, or, considering B as the center of moments, $24 \times$ perpendicular distance $A\ B = 12 \times$ perpendicular distance $B\ C$. In other words, the moment of W about B must equal the moment of P about B—that is, the two moments must be equal. Further, P tends to cause rotation in the direction in which the hands of a watch move, and will, for convenience, be considered positive, or $+$; W tends to cause rotation in a direction opposite to the hands of a watch, and will be considered negative, or $-$. Adding the two algebraically, $P \times B\ C + (-W \times A\ B) = P \times B\ C - W \times A\ B = 0$, since the two moments are equal. Hence the following general

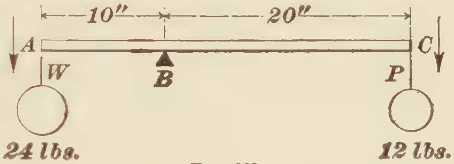


FIG. 322.

Rule.—*One of the necessary conditions of equilibrium is that the* **algebraic** *sum of the moments of all the forces about a given point should equal zero.*

Applications of this rule will occur farther on.

GRAPHICAL EXPRESSIONS FOR MOMENTS.

1372. The moment of a single force may be expressed graphically in the following manner: Let $F = 10$ pounds be the given force (see Fig. 323), and c, at a distance from

$F = f\,c = 7\frac{1}{2}$ feet, be the center of rotation (center of moments). Draw $O\,1$ parallel to F and equal to 10 pounds. Choose any point P as a pole, and draw the rays $P\,O$ and $P\,1$; also draw $P\,2$ perpendicular to $O\,1$. Through any point b on F, draw the sides $a\,b$ and $g\,b$ of the equilibrium polygon; they correspond to $b\,e$ and $d\,e$, Fig. 318, through the intersection of which the resultant must pass, the resultant F being given in the present case. Prolong $a\,b$, and draw

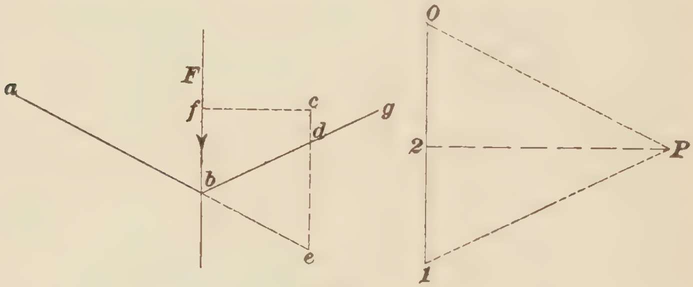


FIG. 323.

$e\,d$ through c parallel to F, intersecting $b\,g$ and $a\,b$ in d and e. It can now be shown that the moment of F about $c = P\,2 \times d\,e$, when $P\,2$ is measured to the same scale as $O\,1$, and $e\,d$ is measured to the same scale as $c\,f$.

The line $P\,2$ is called the **pole distance** and will hereafter be always denoted by the letter H. The line $d\,e$ is called the **intercept.** Hence, the pole distance multiplied by the intercept equals the moment, or, denoting the intercept by y, moment $= H\,y$.

The statement made in the last sentence is one of the most important facts in Graphical Statics, and should be thoroughly understood. In the triangle $P\,O_{1}$, the lines $P\,O$ and P_{1} represent the components of the force F in the directions $a\,b$ and $g\,b$, respectively, while the lines of action of those components are $a\,b$ and $g\,b$, meeting at b. As $d\,e$ is limited by $g\,b$ and $a\,b$ (produced), we may give the following definition: The *intercept* of a force whose moment about a point is to be found is the segment (or portion) which the two components (produced, if necessary) cut off

from a line drawn through the center of moments parallel to the direction of the force.

1373. The pole distance and intercept for the moment of several forces about a given point may be determined in a similar manner, by first finding the magnitude and position of the resultant of all the forces; the moment of this resultant about the given point will equal the value of the resultant moment of all the forces which tend to produce rotation about that point.

EXAMPLE.—Let $F_1 = 20$ pounds, $F_2 = 25$ pounds, and $F_3 = 18$ pounds

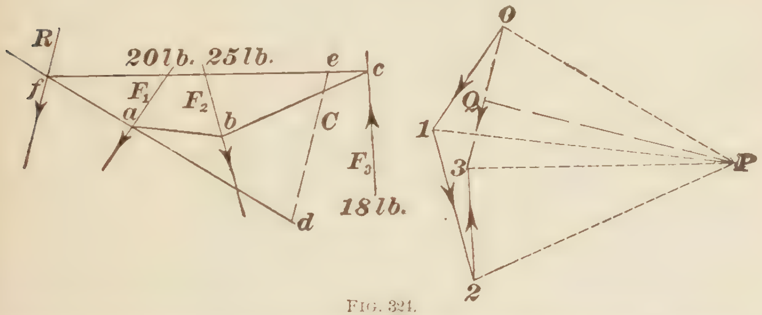


FIG. 324.

be three forces acting as shown in Fig. 324; find their resultant moment about the point C.

SOLUTION.—Draw the force diagram, equilibrium polygon, and resultant R as previously described. Draw $d\,C\,e$ parallel to R. The intercept $d\,e$, multiplied by the pole distance $P\,Q$ = the resultant moment.

1374. If all the forces are parallel, the force polygon is a straight line. This is evident, since if a line of the force polygon be drawn parallel to one of the forces, and from one end of this line a second line be drawn parallel to another force, the second line will coincide with the first.

EXAMPLE.—Let $F_1 = 30$, $F_2 = 20$, and $F_3 = 20$, all in pounds, be three parallel forces acting downwards, as shown in Fig. 325. It is required to find their resultant moment (algebraic sum of the moments) and the moment of their resultant, all moments to be taken about the point C.

SOLUTION.—Lay off $O\,1 = 30$ lb. $= F_1$, $1\text{–}2 = 20$ lb. $= F_2$ and $2\text{–}3 = 20$ lb. $= F_3$, and $O\,3$ is the value of the resultant. Choose some point P as a pole and draw the rays. Take any point, as b, on any force, as F_1, and complete the equilibrium polygon $b\,c\,d\,e\,b$; then, the line of action of the resultant must pass through e. Through C draw $C\,i$ parallel to R and prolong $d\,e$. The moment of R about C equals the pole distance H, multiplied by the intercept $h\,i$, since $h\,i$ is that part of the line drawn through C parallel to R, and included between the lines $b\,e$ and $d\,e$ of the equilibrium polygon which meet upon R. Measuring $h\,i$ to the scale of distances (1 in. = 40 ft.), it equals 23 ft. Measuring H to the scale of forces, it equals 40 lb. Consequently, the moment of R about

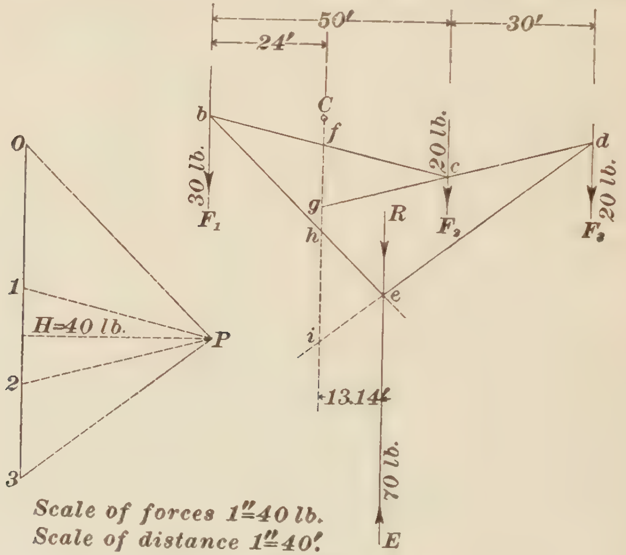


FIG. 325.

$C = 40 \times 23 = 920$ ft.-lb. Considering the force F_3, the intercept is $g\,i$, since F_3 is parallel to R, and, consequently, to $C\,i$; also, $g\,i$ is that part of the line $C\,i$ included between the sides $c\,d$ and $d\,e$, which meet on F_3. Measuring $g\,i$, it is found to equal 28 ft. Hence, the moment of F_3 about $C = 40 \times 28 = 1{,}120$ ft.-lb. The moment of F_2 about $C = H \times f\,g = 40 \times 13 = 520$ ft.-lb. The moment of $F_1 = H \times f\,h = 40 \times 18 = 720$ ft.-lb. Now F_3 and F_2 have positive moments, since they tend to cause rotation in the direction of the hands of a watch, while F_1 has a negative moment, since it tends to cause rotation in the opposite direction. Consequently, adding the moments algebraically, the resultant moment $= 1{,}120 + 520 - 720 = 920$ ft.-lb., the same as the moment of the resultant.

Having described the fundamental principles of Graphical Statics, the subject of Strength of Materials will now be continued, and the stresses due to flexure and torsion discussed.

BEAMS.

1375. Any bar resting upon supports in a horizontal position is called a **beam.**

1376. A **simple beam** is a beam resting upon two supports very near its ends.

1377. A **cantilever** is a beam resting upon one support in its middle, or which has one end fixed (as in a wall) and the other end free.

1378. A **restrained beam** is one which has both ends fixed (as a plate riveted to its supports at both ends).

1379. A **continuous beam** is one which rests upon more than two supports.

In this Course, the continuous beam will not be discussed, as the subject requires a knowledge of higher mathematics.

REACTIONS OF SUPPORTS.

1380. All forces which act upon beams will be considered as vertical, unless distinctly stated otherwise. According to the third law of motion, every action has an equal and opposite reaction. Hence, when a beam is acted upon by downward forces, the supports react upwards. It is required to find the value of the reaction at each support. If a simple beam is uniformly loaded or has a load in the middle, it is evident that the reaction of each support is one-half the load, plus one-half the weight of the beam. If the load is not uniformly distributed over the beam, or if the load or loads are not in the middle, the reactions of the two supports will be different. The upward reactions are considered positive, and the downward forces negative. In

order that a beam may be in equilibrium, three conditions must be fulfilled:

I. The algebraic sum of all vertical forces = 0.

II. The algebraic sum of all horizontal forces = 0.

III. The algebraic sum of the moments of all forces about any point = 0.

Since the loads act downwards and the reactions upwards, the first condition states that the sum of all the loads must equal the sum of the reactions of the supports.

EXAMPLE.—Let R_1 be the reaction of the left support, R_2 the reaction of the right support, and the distance between the two supports 14 feet. Suppose that loads of 50, 80, 100, 70, and 30 pounds are placed

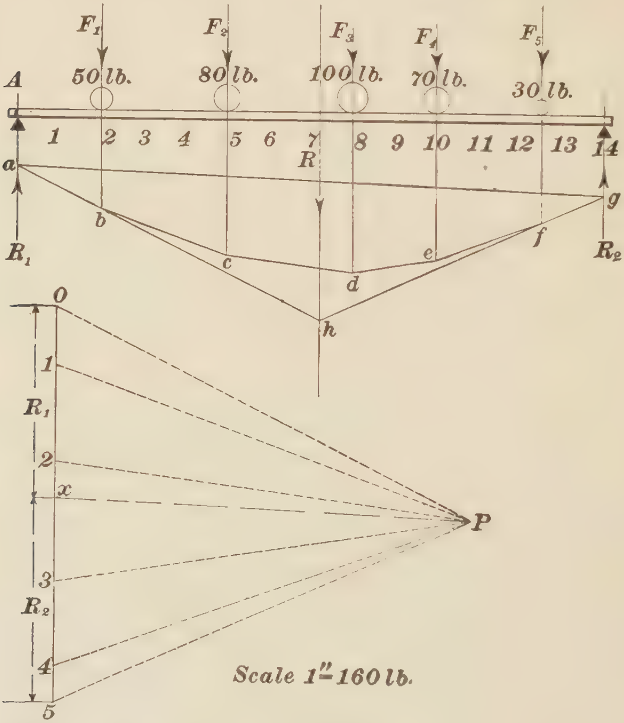


FIG. 326.

on the beam at distances from the left support equal to 2, 5, 8, 10, and $12\frac{1}{2}$ feet, respectively. Required the reactions of the supports, neglecting the weight of the beam. See Fig. 326.

Solution.—The reactions may be found graphically by resolving the resultant of the weights (which, in this case, acts vertically downwards) into two parallel components, passing through the points of support. The sum of the reactions is equal to the sum of the components, but the two sums have different signs. Draw the beam to some convenient scale, and locate the loads as shown in the figure. Draw the force diagram, making $O\ 1 = 50$ lb., $1\text{-}2 = 80$ lb., etc., $O\ 5$ representing the force polygon.

Choose a point b on the line of action of the force F_1, and draw the equilibrium polygon $a\ b\ c\ d\ e\ f\ g\ a$; $a\ b$ and $f\ g$ intersect at h, the point through which the resultant R must pass. Draw $P\ x$ parallel to $a\ g$, and $O\ x$ will be the reaction (= component) R_1, and $x\ 5$ the reaction R_2. Measuring $O\ x$ and $x\ 5$ to the same scale used to draw $O\ 5$, we find $R_1 = 161$ lb., and $R_2 = 169$ lb. By calculation, $R_1 = 161.4$ lb., and $R_2 = 168.6$ lb. This shows that the graphical method is accurate enough for all practical purposes. The larger the scale used, the more accurate will be the results.

The reactions and forces acting upwards will always be considered as positive, or +, and the downward weights as negative, or −. It is plain that the first condition of equilibrium is satisfied when the sum of the positive forces and reactions is equal to the sum of the negative forces.

THE VERTICAL SHEAR.

1381. In Fig. 326, imagine that part of the beam at a minute distance to the left of a vertical line passing through the point of support A, to be acted upon by the reaction $R_1 = 160$ pounds, and that part to the right of the line to be acted upon by the equal downward force due to the load. The two forces acting in opposite directions tend to shear the beam.

Suppose the line had been situated at the point marked *3* instead of at A; the reaction upwards would then be partly counterbalanced by the 50-pound weight, and the total reaction at this point would be $160 - 50 = 110$ pounds. Since the upward reaction must equal the downward load at the same point, the downward force at *3* also equals 110 pounds, and the shear at this point is 110 pounds. At the point *6*, or any point between *5* and *8*, the downward force due to the weight at the left is $50 + 80 = 130$ pounds, and the upward reaction is 160 pounds. The resultant shear is

therefore, 160 − 130 = 30 pounds. At any point between *8* and *10*, the shear is 160 − (50 + 80 + 100) = − 70 pounds. The negative sign means nothing more than that the weights exceed the reaction of the left-hand support.

The vertical shear equals the reaction of the left-hand support, minus all the loads on the beam to the left of the point considered.

For a simple beam, the greatest positive shear is at the left-hand support, and the greatest negative shear is at the right-hand support, and both shears are equal to the reactions at those points.

1382. The vertical shear may be represented graphi-

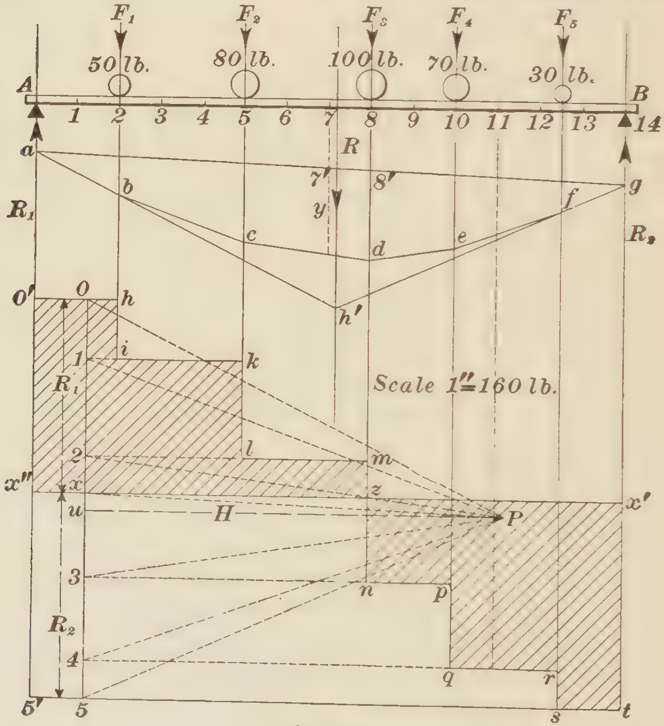


FIG. 327.

cally, as shown in Fig. 327, which is Fig. 326 repeated. Draw the force diagram, continue the lines of action of R_1 and

R_2 downwards and make $O' 5' = O 5$. Through x draw the horizontal line $x x'$, called the **shear axis.** The vertical shear is the same for any point between A and 2 and $= O x = O' x'' = 160$; hence, draw $O' h$ parallel to $x x'$, and any ordinate measured from $x x'$ to $O' h$, between O' and $h = 160$ pounds $=$ the vertical shear at any point between O' and h when $O' h = A 2$, Through 1, draw $1 k$ and project the points 2 and 5 upon it, in i and k. Then, the length of the ordinate between $x x'$ and $i k =$ the vertical shear between 2 and 5. In the same way, find the remaining points l, m, etc. The broken line $O' h i$..................t is called the **shear line;** and the figure $O' h i$..................$r s t x' x''$ is called the **shear diagram.** To find the shear at any point, as 11, project the point upon the shear axis and measure the ordinate to the shear line, drawn through the projected point. If the ordinate is measured from the shear axis upwards, the vertical shear is positive; if downwards, it is negative. For the point 11, the vertical shear $= -140$ lb. The maximum negative vertical shear is -170 lb. $= x' t = x\ 5$. The greatest shear, whether positive or negative, is the one which the beam must be designed to withstand.

1383. A beam seldom fails through shearing, but generally breaks by reason of the load bending and breaking it; that is, through flexure. In order to design a beam to resist flexure, the greatest (maximum) bending moment must be known.

The ***bending moment*** *at any point of a beam is the algebraic sum of the moments of all of the forces (the reaction included) acting upon the beam, on either side of that point, the point being considered as the center of moments.*

The expression "algebraic sum" refers to the fact that, when considering the forces acting at the left of the point taken, the moments of all the forces acting upwards are considered positive, and the moments of all the forces acting downwards are considered negative. Hence, the algebraic sum is the moment of the left reaction about the given point, minus the sum of the moments about the same point of all

the downward forces between the reaction and the given point. Should there be any force or forces acting upwards, their moments must be *added*, since they are positive. If the forces on the right of the point are considered, all lever arms are negative, distances to the left of the point being +, to the right, − (see Art. **1371**). Hence, the downward forces give positive moments, and the right reaction gives a negative moment. This is as it should be, for the downward forces on the right and the upward forces on the left tend to rotate the beam in the direction of the hands of a watch, while the downward forces on the left and the upward forces on the right tend to rotate the beam in the opposite direction.

1384. To find the bending moment for any point of a beam, as 7 in Fig. 327, by the graphical method, draw the vertical line 7-7′ through the point. Let y = that part of the line included between $a\ g$ and $a\ c\ f\ g$ of the equilibrium polygon (= vertical intercept). Then, $H \times y$ = the bending moment. H, of course, = the pole distance = $P\ u$. For any other point on the beam, the bending moment is found in the same manner—i. e., by drawing a vertical line through the point and measuring that part of it included between the upper and lower lines of the equilibrium polygon. The scale to which the intercept y is measured is the same as that used in drawing the length $A\ B$ of the beam. The pole distance H is measured to the same scale as $O\ 5$. In the present case, $y = 2.05$ feet, and $H = 349$ pounds; hence, the bending moment for the point 7 is $H\ y = 349 \times 2.05 = 715.45$ foot-pounds.

NOTE.—The expression "foot-pounds," used in stating the value of a moment, must not be confounded with foot-pounds of work. The former means simply that a force has been multiplied by a distance, while the latter means that a resistance has been overcome through a distance. In expressing the value of a moment, the force is usually measured in pounds or tons, and the distance in inches or feet; hence, the moment may be inch-pounds or inch-tons and foot-pounds or foot-tons. Unless otherwise stated, the bending moment will always be expressed in inch-pounds, the length of the beam being always measured in inches, and, consequently, also the length of the intercept y.

1385. If expressed in inch-pounds, the value of the moment just found is $715.45 \times 12 = 8{,}585.4$ inch-pounds.

It will be noticed that after the force diagram and equilibrium polygon have been drawn the value of the bending moment depends solely upon the value of y, since the length $P u = H$ is fixed. At the points a and g, directly under the points of support of the beam, $y = 0$; hence, for these two points, bending moment $= H y = H \times 0 = 0$; that is, for any simple beam, the bending moment at either support is zero. The greatest value for the bending moment will evidently be at the point 8, since $d\,8'$ is the longest vertical line which can be included between $a\,g$ and $a\,c\,f\,g$.

The figure $a\,c\,e\,g\,a$ is called the **diagram of bending moments.**

1386. Consider now the case of a simple beam uniformly loaded. Let the distance between the supports in Fig. 328 be 12 feet, and let the total load uniformly distributed over the beam be 216 pounds. Divide the load into a convenient number of equal parts, the more the better, say 12, in this case. The load which each part represents is $216 \div 12 = 18$ pounds. For convenience, lay off $O\,C$ on the vertical through the left-hand support, equal to 216 pounds to the scale chosen, and divide it into 12 equal parts, $O\,a$, $a\,b$, etc.; each part will represent 18 pounds to the same scale. Choose a pole P, and draw the rays $P\,O$, $P\,a$, $P\,b$, etc. Through the points d, e, etc., the centers of gravity of the equal subdivisions of the load, draw the verticals $d\,1$, $e\,3$, $f\,5$, etc., intersecting the horizontals through O, a, b, etc., in 1, 3, 5, etc. Draw $O\,1$, 1-2, 2-3, 3-4, 4-5, etc., and the broken line thus found will be the shear line. In drawing the shear line for a uniform load in this manner, it is assumed that each part of the total load is concentrated at its center of gravity, or, in other words, that a force equal to each small load (18 lb.) acts upon the beam at each of the points d, e, f, etc.

Construct the diagram of bending moments in the ordinary manner by drawing $g\,i$ parallel to $P\,O$, $i\,k$ parallel to $P\,a$, etc. Draw $P\,M$ parallel to $g\,h$, and $M\,q$ horizontal; $M\,q$ is the shear axis. When the load is uniform and the

work has been done correctly, $O\ M$ should equal $M\ C$—that is, the reactions of the two supports are equal.

1387. The shear line is not a broken line in reality, as shown, since the load is distributed evenly over the entire beam, and not divided into small loads concentrated at d, e, f, etc., as was assumed. The points *1*, *3*, etc., are evidently

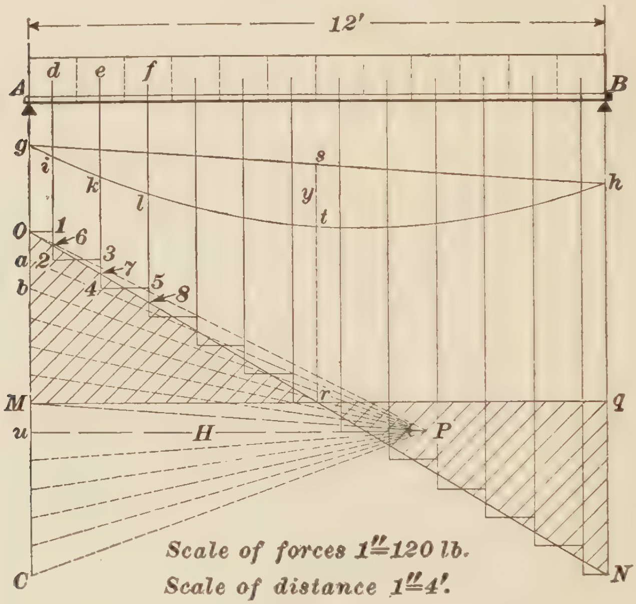


FIG. 328.

too high, and the points *2*, *4*, etc., too low. To find the real shear line, bisect the lines *1-2*, *3-4*, etc., locating the points *6*, *7*, *8*, etc. Trace a line through O, *6*, *7*, *8*..........N, and it will be the real shear line. For all cases of a uniform load, the shear line will be straight and may be drawn from O to N directly.

The diagram of bending moments is also not quite exact, but may be corrected by tracing a curve through the points g and h, which shall be tangent to $g\ i$, $i\ k$, $k\ l$, etc., at their middle points, as shown in the figure.

1388. To find the maximum bending moment for any beam having two supports, draw a vertical line through r, where the shear line cuts the shear axis, and the intercept, $s\,t = y$, on the diagram of bending moments, will be the greatest value of y, and, consequently, the greatest bending moment $= H \times y$. In the present example, $H = P\,u = 247.5$ pounds and maximum $y = s\,t = 15.72$ inches; hence, the maximum bending moment $= H\,y = 247.5 \times 15.72 = 3,890.7$ inch-pounds. The above is true, no matter how the beam may be loaded. In Fig. 327, the shear line cuts the shear axis at z, and $d\,8'$, on the vertical through z, was shown previously to be the maximum intercept.

1389. If there is a uniform load on the beam and one or more concentrated loads, as in Fig. 329, the method of finding the moment diagram and shear line is similar to that used in the last case. In Fig. 329, let the length $A\,B$ of the beam be 15 feet; the uniformly distributed load 180 pounds, with two concentrated loads, one of 24 pounds, 5 feet from A, and the other of 30 pounds, 11 feet from A. Draw the beam and loads as shown, the length of the beam and the distances of the weights from A being drawn to scale. Divide the uniform load into a convenient number of equal parts, say 10 in this case; each part will then represent $\frac{180}{10} = 18$ pounds. Draw $A\,O\,C$, as usual, and lay off 3 of the 18-pound subdivisions from O downwards; then lay off 24 pounds, to represent the first weight. Lay off four more of the equal subdivisions, and then the 30-pound weight. Finally, lay off the remaining three equal subdivisions, the point C being the end of the last 18-pound subdivision. $O\,C$ should then equal $180 + 24 + 30 = 234$ pounds to the scale to which the weights were laid off. It will be noticed in the above that the equal subdivisions of the load were laid off on $O\,C$ until that one was reached on which the concentrated loads rested, and that the concentrated loads were laid off before the equal subdivision on which the concentrated load rested was laid off. Had one of the concentrated loads been to the right of the center of gravity of the subdivision on which it

rests, the weight of the subdivision would have been laid off first. Locate the centers of gravity of the equal subdivisions and draw the verticals through them as in the previous case. Choose a pole P, draw the rays, the diagram of bending moments, and the shear axis $M\,q$, as previously described.

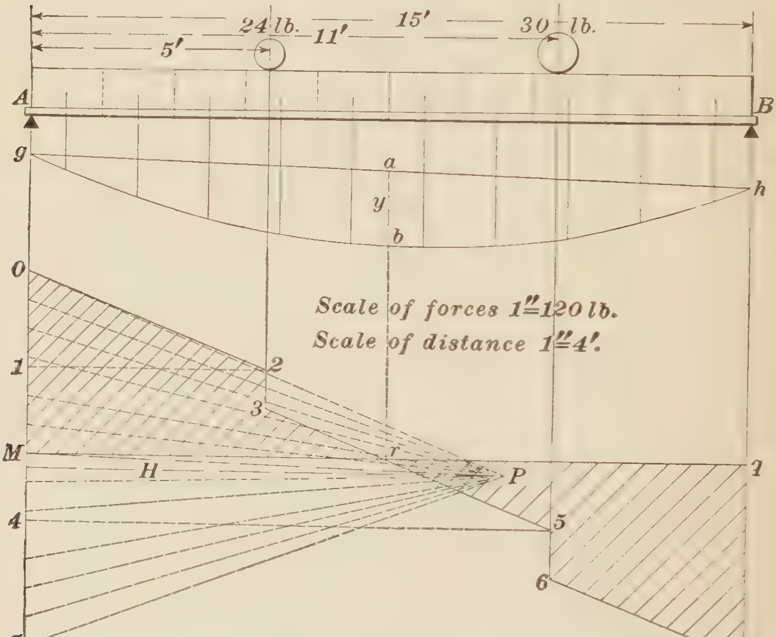


Fig. 329.

To find the maximum bending moment, the shear line must be drawn and its intersection with the shear axis determined.

The weight of the uniform load per foot of length is $\frac{180}{15} = 12$ lb. The weight of that part between A and the center of the 24-lb. weight is $12 \times 5 = 60$ lb. Lay off $O\,1 = 60$ lb. and draw 1-2 horizontal, cutting the vertical through the center of the 24-lb. weight in 2. Draw $O\,2$ and it will be a part of the shear line. Lay off 2-3 vertically downwards, equal to 24 lb., locating the point 3. Lay off $O\,4$ equal to $12 \times 11 + 24 = 156$ lb., and draw the horizontal 4-5, cutting the vertical through the center of the 30-lb.

weight in *5*. Join *3* and *5* by the straight line *3-5*. Lay off *5-6* vertically downwards equal to 30 lb. Draw the horizontal $C\,N$, intersecting the vertical $B\,q\,N$ in N and join *6* and N by the straight line $6\,N$. The broken line O *2-3-5-6* N is the shear line and cuts the shear axis in the point r. Drawing the vertical $r\,b\,a$, through r, it intersects the moment diagram in a and b; hence, $a\,b$ is the maximum y. For this case, the maximum bending moment $= H \times y = 300 \times 18.36 = 5{,}508$ in.-lb.

It is better, in ordinary practice, to choose the pole P on a line perpendicular to $O\,C$ and at some distance from $O\,C$ easily measured with the scale used to lay off $O\,C$. Thus, suppose $O\,C$ to be laid off to a scale of $1'' = 60$ lb. At some convenient point, as M, on $O\,C$, draw a perpendicular line and choose a point on this line whose distance from $O\,C$ shall be easily measurable, say $3\frac{1}{2}''$. Then, H is known to be exactly $60 \times 3\frac{1}{2} = 210$ lb. and will not have to be measured when finding the bending moment $H\,y$.

1390. If the student has familiarized himself with the method of constructing the shear and moment diagrams for concentrated loads, he will find no difficulty in understanding the preceding operations, which may be condensed into the following

Rule.—*Divide the beam into an even number of parts* (the greater the better), *and the uniform load into half as many. Consider the divisions of the load as concentrated loads applied, alternately, at the various points of division of the beam* (the ends included); *that is, the first point of division* (*the support*) *carries no load, the next one does, the following one does not*, etc. *Then proceed as in the ordinary case of concentrated loads.*

EXAMPLE.—Find the reactions of the supports, the maximum bending moments, and the maximum vertical shear of the beam shown in Fig. 330, which has one overhanging end.

SOLUTION.—Draw $O\,C$ and the force diagram in the usual manner. Construct the bottom curve of the moment diagram in the same manner as in the preceding cases. The side $d\,e$ is parallel to $P\,4$; $e\,h$ is parallel to $P\,C$, and cuts the vertical through the right reaction in h.

Join h and g by the straight line $g\,h$, and draw $P\,M$ parallel to $g\,h$. Then, $O\,M = 77$ lb. = left reaction and $M\,C = 173$ lb. = right reaction. The shear line is drawn as in the previous cases until the point n, on the vertical $h\,n$, is reached; $k\,n$ here denotes the vertical shear for any point between the 50-lb. weight and the right support, and this shear is negative. The point k denotes the intersection of the shear axis and the vertical through the right support. For any point to the

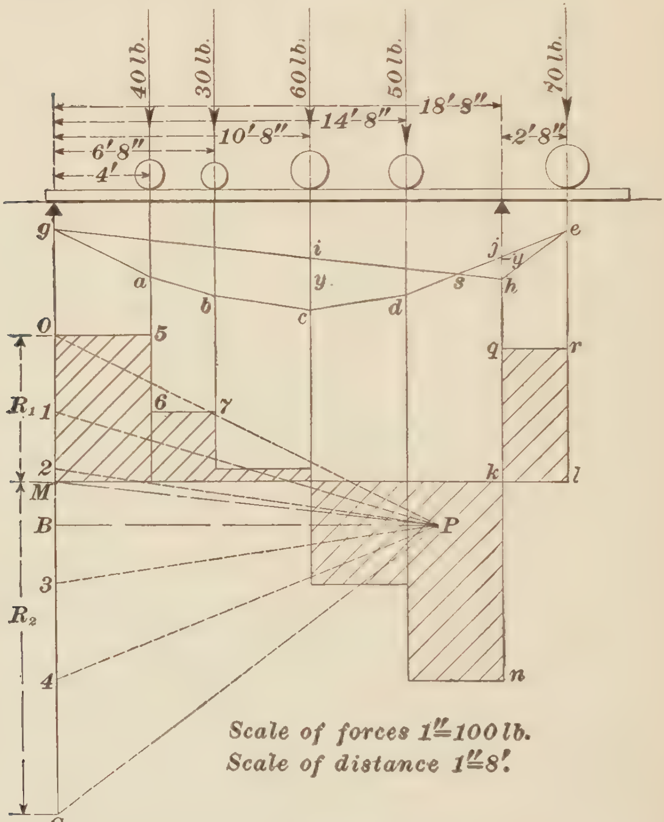


FIG. 330.

right of k, between k and l, the vertical shear is positive, and is equal to 70 lb.; hence, lay off $k\,q$ upwards equal to 70 lb., and draw $q\,r$ horizontal. The line, O 5-6-7 ―――― $n\,q\,r$ is the shear line. Measuring $O\,M$, $k\,n$, and $k\,q$, it is found that $O\,M = 77$ lb., $k\,n = -103$ lb., and $k\,q = 70$ lb.; therefore, $k\,n = -103$ lb. = maximum vertical shear; $c\,i$ is evidently the maximum y; hence, the maximum bending moment =

$Hy = PB \times ci = 200 \times 26'' = 5{,}200$ in.-lb. Any value of y measured in the polygon $g\,a\,b\,c\,d\,s\,g$ is positive, and any value measured in the triangle $e\,h\,s$ is negative. Consequently, the bending moment for any point between s and the vertical, through the center of the 70-lb. weight, is negative, since $H \times (-y) = -Hy$. In all cases, when design-

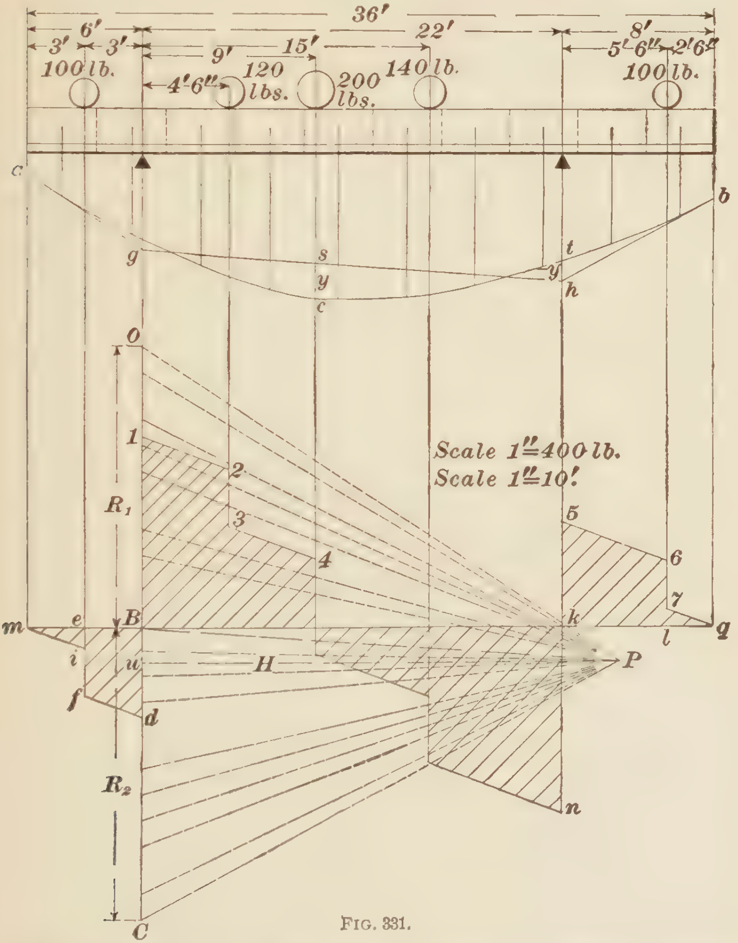


FIG. 331.

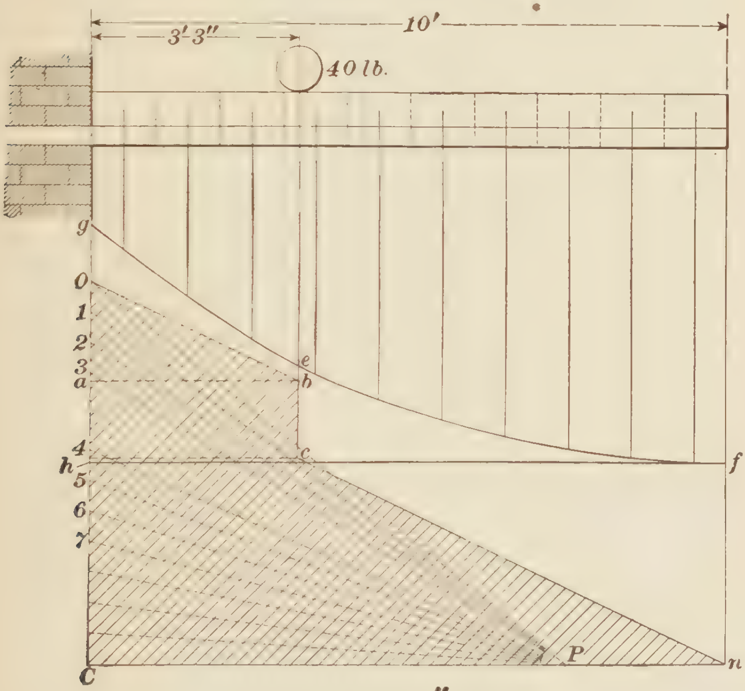
ing beams with overhanging studs, the maximum bending moment, whether positive or negative, should be used. In the present case, the maximum negative y, or $h\,j$, is less than the maximum positive y, or $c\,i$; therefore, the maximum negative bending moment is also less than the maximum positive moment.

EXAMPLE.—Fig. 331 shows a beam overhanging both supports, which carries a uniform load of 15 pounds per foot of length, and has five concentrated loads at distances from the supports as marked in the figure. Required the reactions of the supports, the maximum positive and negative bending moments, and the maximum vertical shear.

SOLUTION.—Construct the force diagram and equilibrium polygon in the ordinary manner, continuing the latter to b and a, points on the verticals passing through the ends of the beam. Draw $b\,h$ and $a\,g$ parallel to $P\,C$ and $P\,O$ respectively, intersecting the verticals through the points of support in h and g. Join g and h, and draw $P\,B$ parallel to $g\,h$. Then, $O\,B = 588$ lb. $= R_1$, and $B\,C = 612$ lb. $= R_2$. Through B, draw the shear axis $m\,q$. To draw the shear line, proceed as follows: The shear for any point to the left of the left support is negative, and for any point to the right of the right support is positive; between the two supports it is positive or negative, according to the manner of loading, and the point considered. The negative shear at the left support $= 15 \times 6 + 100 = 190$ lb.; hence, lay off $B\,d$ downwards equal to 190 lb. For a point to a minute distance to the right of e, the shear is $15 \times 3 + 100 = 145$ lb. $= e\,f$, and for a minute distance to the left, it is $15 \times 3 = 45$ lb. $= e\,i$; at m, it is 0. Consequently, $m\,i\,f\,d$ is the shear line between the end of the beam and the left support. Lay off $O\,1 = B\,d = 190$ lb. and draw the shear line *1-2-3-4*———n in the usual manner. Draw the shear line *5-6-7* q, laying off $k\,5 = 15 \times 8 + 100 = 220$ lb.; $l\,6 = 100 + 15 \times 2\frac{1}{2} = 137\frac{1}{2}$ lb., and $6\text{-}7 = 100$ lb. At q, the vertical shear is again 0. The broken line $m\,i\,f\,d$ *1-2-3*———n *5-6-7* q is the shear line. The maximum positive bending moment is $H \times y = P\,u \times s\,c = 1{,}000 \times 22.5 = 22{,}500$ in.-lb. The greatest maximum negative moment is $H \times (-y) = P\,u \times -t\,h = 1{,}000 \times -11.6 = -11{,}600$ in.-lb. It will be noticed that there are two negative and one positive maximum bending moments.

1391. The student should now be able to find the bending moment for any beam having but two supports, whatever the character of the loading. The bending moment plays a very important part in the flexure of beams, which is the next subject to be considered. In all cases of loading heretofore considered, no other forces than the loads themselves have been considered. Should forces act upon the beam which are not vertical, the force polygon will be no longer a straight line, but a broken line somewhat similar in character to O *1-2-3-4-5* in Fig. 321.

In Fig. 332 is shown a cantilever beam projecting 10 ft. from the wall. It carries a uniform load of 16 pounds per foot of length, and a concentrated load of 40 pounds at a distance of $3\frac{1}{4}$ feet from the wall. The maximum bending moment is required. The method is similar to the last, except that, as there is but one support, there can be but one reaction. Since the beam is 10 feet long, the total



Scale of forces $1'' = 100$ *lb.*
Scale of distance $1'' = 3'$.

FIG. 332.

weight of the uniform load is $16 \times 10 = 160$ pounds. Hence, the reaction $= 160 + 40 = 200$ pounds.

Draw $O\,C$ equal to 200 lb., to some convenient scale. Draw $P\,C$ perpendicular to $O\,C$ at C and choose the pole P at a convenient distance from $O\,C$. For convenience, divide the uniform load into 10 equal parts, as shown; then,

each part will represent 16 lb. Lay off *O 1*, *1-2*, *2-3*, each equal to 16 lb., and *3-4* equal to 40 lb. Also, *4-5*, *5-6*, *6-7*, etc., equal to 16 lb. each. 3 ft. 3 in. $= 3\frac{1}{4}$ ft., and $16 \times 3\frac{1}{4} = 52$ lb. Lay off $O\,a = 52$ lb., and draw *a b*, meeting the vertical through the center of the weight in *b*. Draw *O b*; lay off *b c* equal to 40 lb. and draw *C n*. *O b c n* is the shear line. The perpendicular through the point *C* coincides with *P C*; hence, *n P C* is the shear axis. Draw the line *g c f* of the moment diagram as in the previous cases. In Fig. 329, and in the preceding figures, the line *g h* was drawn connecting the extreme ends of the bottom line; in other words, it joined the points where the equilibrium polygon cut the lines of direction of the reactions of the supports. This cannot be done in this case, because there is no right reaction; therefore, *g h* must be drawn by means of some other property of the polygon. In the previous cases, the shear axis was drawn perpendicular to *O C* at the point where a line through the pole *P* parallel to *g h* cut *O C*, or, in other words, the shear axis was drawn through the point which marked the end *M* of the left reaction *O M*. In the present case, Fig. 332, the point *C* is the end of the left reaction; hence, *f h* is parallel to *P C*, *h g* is the maximum *y*, and the bending moment $= H\,y = P\,C \times h\,g = 250 \times 45 =$ 11,250 inch-pounds.

It will also be noticed that the distance $y = h\,g$ is measured from the line *h f* upwards, while, for all points between the supports in the previous examples, this distance was measured downwards. The same observation is true for any point between *h* and *f*; hence, for a cantilever, all bending moments are negative.

1392. All the cases of beams heretofore given might have been solved by analytical methods—that is, by algebraic processes and formulas; but the graphical method is to be preferred, as it is usually shorter, very nearly as accurate, and less liable to error. Moreover, when the diagram has once been drawn, both the bending moment and the shear for any point can be instantly determined. Ex-

cept in special cases, the graphical method will be employed to determine the maximum bending moment, but in some particular cases formulas will be used, by which the result can be obtained more quickly than by the graphical method.

EXAMPLES FOR PRACTICE.

1. A simple beam 24 feet long carries 4 concentrated loads of 160, 180, 240, and 120 pounds at distances from the left support of 4, 10, 16, and 21 feet, respectively. (*a*) What are the values of the reactions? (*b*) What is the maximum bending moment in inch-pounds?

Ans. $\begin{cases} (a)\ R_1 = 333\frac{1}{3} \text{ lb.};\ R_2 = 366\frac{2}{3} \text{ lb.} \\ (b)\ 28{,}480 \text{ in.-lb.} \end{cases}$

2. A simple beam carries a uniform load of 40 pounds per foot, and supports two concentrated loads of 500 and 400 pounds at distances from the left support of 5 and 12 feet, respectively. The length of the beam is 18 feet. What are (*a*) the reactions? (*b*) The maximum bending moment in inch-pounds?

Ans. $\begin{cases} (a)\ R_1 = 854\frac{4}{9} \text{ lb.};\ R_2 = 765\frac{5}{9} \text{ lb.} \\ (b)\ 48{,}846 \text{ in.-lb.} \end{cases}$

3. A cantilever projects 10 feet from a wall and carries a uniform load of 60 pounds per foot; it also supports three concentrated loads of 100, 300, and 500 pounds at distances from the wall of 2, 5, and 9 feet, respectively. Required, (*a*) the maximum vertical shear, and (*b*) the maximum bending moment in inch-pounds.

Ans. $\begin{cases} (a)\ -1{,}500 \text{ lb.} \\ (b)\ -110{,}400 \text{ in.-lb.} \end{cases}$

4. A beam which overhangs one support sustains six concentrated loads of 160 lb. each at distances from the left support of 4 ft. 9 in., 7 ft., 9 ft. 6 in., 12 ft., 15 ft., and 18 ft. 3 in., respectively, the distance between the supports being 16 ft. What are (*a*) the reactions? (*b*) The maximum bending moment?

Ans. $\begin{cases} (a)\ R_1 = 295 \text{ lb.};\ R_2 = 665 \text{ lb.} \\ (b)\ 20{,}460 \text{ in.-lb.} \end{cases}$

5. A beam which overhangs both supports equally carries a uniform load of 80 pounds per foot, and has a load of 1,000 pounds in the middle, the length of the beam being 15 feet, and the distance between the supports 8 feet. What is (*a*) the vertical shear? (*b*) The maximum bending moment?

Ans. $\begin{cases} (a)\ 820 \text{ lb.} \\ (b)\ 25{,}800 \text{ in.-lb.} \end{cases}$

NOTE.—The student may not obtain the exact answers given above, but if his results do not differ by more than 1%, he will know that his method is right.

NEUTRAL AXIS.

1393. In Fig. 333, let *A B C D* represent a cantilever. Suppose that a force *F* acts upon it at its extremity *A*. The beam will then be bent into the shape shown by *A' B C D'*.

It is evident from the cut that the upper part $A' B$ is now longer than it was before the force was applied; i. e., $A' B$ is longer than $A B$. It is also evident that $D' C$ is shorter than $D C$. Hence, the effect of the force F in bending the beam is to lengthen the upper fibers and to shorten

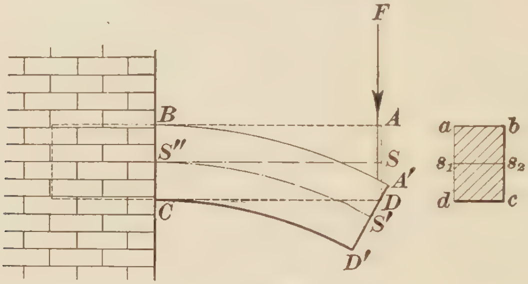


FIG. 333.

the lower ones. In other words, when a cantilever is bent through the action of a load, the upper fibers are in tension and the lower fibers in compression. The reverse is the case in a simple beam in which the upper fibers are in compression and the lower fibers in tension. Further consider-

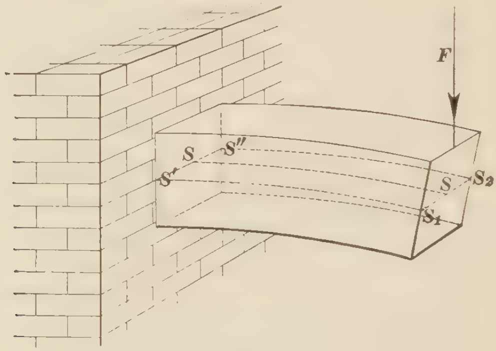


FIG. 334.

ation will show that there must be a fiber, $S\ S''$, which is neither lengthened nor shortened when the beam is bent, i. e., $S'\ S'' = S\ S''$. When the beam is straight the fiber $S\ S''$, which is neither lengthened nor shortened when the beam is bent, is called the **neutral line.** There may be

any number of neutral lines dependent only on the width of the beam. For, let $b\ a\ d\ c$, Fig. 333, be a cross-section of the beam. Project s upon it in s_1. Make $b\ s_2 = a\ s_1$ and draw $s_1\ s_2$; then, any line in the beam which touches $s_1\ s_2$ and is parallel to $S\ S''$ is a neutral line. Thus, in Fig. 334, $S_1\ S'$, $S\ S$, $S_2\ S''$, etc., are all neutral lines. The line $S_1\ S_2$ is called the **neutral axis,** and the surface $S_1\ S'\ S''\ S_2$ is called the **neutral surface**. The neutral axis, then, is the line of intersection of a cross-section with the neutral surface. It is shown in works on mechanics that *the neutral axis always passes through the center of gravity of the cross-section of the beam.*

1394. Experimental Law.—*When a beam is bent, the horizontal elongation (or compression) of any fiber is directly proportional to its distance from the neutral surface, and, since the strains are directly proportional to the horizontal stresses in each fiber, they are also directly proportional to their distances from the neutral surface, provided the elastic limit is not exceeded.*

1395. Suppose the beam to be a rectangular prism, then every cross-section will be a rectangle, and the neutral axis will pass through the center o. See Fig. 335.

Let the perpendicular distance from the neutral axis $M\ N$ to the *outermost fiber* be denoted by c, and the horizontal unit stress (stress per square inch) at the distance c from the axis by S. If a is the area of a fiber, the stress on the outermost fibers will be $a\,S$. The stress on a fiber at the distance unity (1 inch) from $M N$ is $\frac{a\,S}{c}$; and the stress on a fiber at the distance r_1 is $\frac{a\,S}{c} \times r_1 = \frac{a\,S\,r_1}{c}$. The moment of this stress about the axis MN is $\frac{a\,S\,r_1}{c} \times r_1 = \frac{a\,S\,r_1^2}{c} = \frac{S}{c}\,a\,r_1^2$. The moment of the

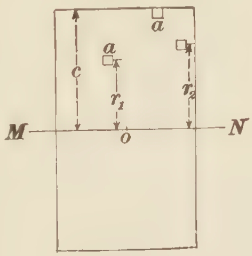


FIG. 335.

stress on any other fiber at a distance r_2 from MN is evidently $\frac{S}{c} a r_2^2$, and for a distance r_3, $\frac{S}{c} a r_3^2$, etc. If n is the number of fibers, the sum M of the moments of the horizontal stresses on all the fibers is

$$M = \frac{S}{c} a r_1^2 + \frac{S}{c} a r_2^2 + \frac{S}{c} a r_3^2 +, \text{etc.}, =$$

$$\frac{S}{c}(a r_1^2 + a r_2^2 + a r_3^2 + \ldots\ldots a r_n^2) = \frac{S}{c} a (r_1^2 + r_2^2 + r_3^2 + \ldots\ldots r_n^2).$$

Now, let r be a quantity whose square equals the mean of the squares of $r_1, r_2, r_3, \ldots\ldots r_n$. Then, $r^2 = \frac{r_1^2 + r_2^2 + \ldots\ldots + r_n^2}{n}$; and, therefore, $r_1^2 + r_2^2 + r_3^2 + \ldots\ldots + r_n^2 = n r^2$. Substituting above, we get $M = \frac{S}{c} n a r^2$. But, since a is the area of one fiber, $n a$ is the area of all the fibers—that is, the area A of the cross-section; hence, the sum of the moments of all the horizontal stresses $= \frac{S}{c} A r^2$.

1396. The expression $A r^2$, which is found by dividing a section into a large number of minute areas (a, a, etc.), multiplying each area by the square of its distance from an axis (r_1^2, r_2^2, r_3^2, etc.), and then adding the products thus obtained, is called the **moment of inertia** of the section with respect to that axis, and is usually denoted by the letter I. Hence,

$$I = A r^2. \qquad \textbf{(115.)}$$

1397. The quantity r, whose square is the mean of the squares of all the distances of the minute areas from the axis, is called the **radius of gyration.**

1398. The sum of the moments of all the horizontal stresses may then be written as $\frac{S}{c} A r^2 = \frac{S}{c} I$, or $S \frac{I}{c}$. This expression is called the **resisting moment,** since it is the measure of the resistance of the beam to bending (and, consequently, to breaking) when loaded. The resisting moment must equal the bending moment when the beam

is in equilibrium; hence, denoting the bending moment by M,

$$M = S_4 \frac{I}{c}. \qquad \textbf{(116.)}$$

1399. The values of I and c depend wholly upon the size and form of the cross-section of the beam, and S_4 is the ultimate strength of flexure of the material.

In Table 29, the average ultimate strength of flexure S_4 is given for a number of different materials.

TABLE 29.

Material.	Ultimate Strength of Flexure in Lb. per Sq. In. S_4.
Cast Iron	38,000
Wrought Iron	45,000
Steel	120,000
Brass	17,000
Ash	14,000
Brick	1,000
Stone	2,000
Hemlock	7,200
Oak, white	12,500
Pine, white	9,000
Pine, yellow	11,000
Hickory	16,000

1400. Exact values of I for most cross-sections can only be determined by the aid of the calculus. The least value of I occurs when the axis passes through the center of gravity of the cross-section—that is, when I is found with reference to the neutral axis.

The least moments of inertia for a number of different sections are given in the Table of Moments of Inertia; also, the area of the sections and the values of c. The dotted line indicates the position of the neutral axis, about which the moment of inertia is taken.

In the Table of Moments of Inertia, A is the area of the section, and π = ratio of the circumference of a circle to its diameter = 3.1416. It will be noticed that d is always taken vertically.

1401. To use formula **116**, find the bending moment in inch-pounds by the graphical method previously described, or calculate it by means of the Table of Bending Moments. If it is desired to find the size of a beam that will safely resist a given bending moment, take S_4 from Table 29, Art. **1399**, and divide it by the proper factor of safety taken from Table 28. Then, formula **116** becomes

$$M = \frac{S_4 I}{f c}. \qquad \textbf{(117.)}$$

From this $\frac{I}{c} = \frac{M f}{S_4}$. Substituting the values of M, f, and S_4, the value of $\frac{I}{c}$ is found. The kind and shape of beam having been decided upon, the size can be so proportioned that $\frac{I}{c}$ for the section shall not be less than the value calculated above. An example will make this clear.

EXAMPLE.—What should be the size of an ash girder to resist safely a bending moment of 28,000 inch-pounds, the cross-section to be rectangular and the load steady?

SOLUTION.—$M = \frac{S_4 I}{f c}$, or $\frac{I}{c} = \frac{M f}{S_4}$.

$M = 28{,}000$; from Table 29, Art. **1399**, $S_4 = 14{,}000$; from Table 28, Art. **1362**, $f = 8$.

Then, $\frac{I}{c} = \frac{28{,}000 \times 8}{14{,}000} = 16.$

From the table of Moments of Inertia, $I = \frac{b d^3}{12}$ and $c = \frac{d}{2}$ for a rectangle; hence, $\frac{I}{c} = \frac{b d^3}{12} \times \frac{2}{d} = \frac{b d^2}{6} = 16$, or $b d^2 = 96$. Any number of values of b and d can be found that will satisfy this equation. If b is taken as 6 inches, $d^2 = \frac{96}{6} = 16$ and $d = \sqrt{16} = 4$. Hence, the beam may be a 6×4, with the short side vertical. When possible, it is always better to have the longer side vertical. If b is taken as 2 inches, $d^2 = 48$ and $d = \sqrt{48} = 7$ inches, nearly; hence, a 2×7 will also

answer the purpose. The advantage of using a 2×7 instead of a 6×4 is evident, since the 6×4 contains nearly twice as much material as the 2×7. Thus, the area of the cross-section of a 6×4 is 24 square inches, and of a 2×7, 14 square inches. Moreover, the 2×7, with its long side vertical, is slightly stronger than the 6×4, with its short side vertical, since $\frac{I}{c} = \frac{2 \times 7^2}{6} = 16\frac{1}{3}$ for the former, and $\frac{6 \times 4^2}{6} = 16$ for the latter. If the 6×4 had its longer side vertical, thus making it a 4×6, $\frac{I}{c}$ would then equal $\frac{4 \times 6^2}{6} = 24$, and the safe bending moment could be increased to $M = \frac{S_4 I}{f c} = \frac{14,000 \times 24}{8} = 42,000$ in.-lb.

1402. If the breaking bending moment, form and size of the cross-section of the beam are known, the ultimate strength of flexure S_4 can be readily found from formula **116**, by substituting the values of M, I, and c, and solving for S_4.

EXAMPLE.—A cast iron bar, 2 inches square, breaks when the maximum bending moment = 63,360 inch-pounds; what is its ultimate strength of flexure?

SOLUTION.—$M = S_4 \frac{I}{c}$, or $S_4 = \frac{M c}{I}$. $c = \frac{d}{2} = 1''$.

$I = \frac{d^4}{12} = \frac{2^4}{12}$; therefore, $S_4 = \frac{63,360 \times 1}{\frac{1}{12} \times 2^4} = 47,520$ lb. per sq. in.

1403. In order to save time in calculating, the bending moments for cases of simple loading are given in the Table of Bending Moments. W denotes a concentrated load, and w the uniform load per inch of length. All dimensions are to be taken in inches when using the formulas.

For any other manner of loading than is described in the Table of Bending Moments, the maximum bending moment must be found by the graphical method.

EXAMPLE.—A wrought iron cantilever, 6 feet long, carries a uniform load of 50 pounds per inch. The cross-section of the beam is an equilateral triangle, with the vertex downwards; what should be the length of a side?

SOLUTION.—$M = \frac{w l^2}{2}$, from the Table of Bending Moments, $= \frac{50 \times (6 \times 12)^2}{2} = 129,600$ in.-lb. $I = \frac{b d^3}{36}$ and $c = \frac{2}{3} d$, from the Table

of Moments of Inertia; hence, $\frac{I}{c} = \frac{b\,d^2}{24}$. $S_4 = 45{,}000$, from Table 29, Art. **1399,** and $f = 4$, from Table 28, Art. **1362.** Therefore, $129{,}600 = \frac{45{,}000}{4} \times \frac{b\,d^2}{24}$, or $b\,d^2 = \frac{129{,}600 \times 4 \times 24}{45{,}000} = 276.48$. Since an equilateral triangle has been specified, b can not be given *any* convenient value in order to find d. For an equilateral triangle, $d = b \sin 60° = .866\,b$. Hence, $b\,d^2 = b\,(.866\,b)^2 = .75\,b^3$. Therefore, $b\,d^2 = .75\,b^3 = 276.48$ or $b = \sqrt[3]{\frac{276.48}{.75}}$ $\log b = \frac{\log 276.48 - \log .75}{3} = .85554$, or $b = 7.17''$, nearly.

EXAMPLE.—What weight would be required to break a round steel bar 4 inches in diameter, 16 feet long, fixed at both ends and loaded in the middle?

SOLUTION.—Use formula **116;** $M = \frac{S_4\,I}{c}$. Here $M = \frac{W\,l}{8}$, from the Table of Bending Moments, Vol. V; $S_4 = 120{,}000$; $\frac{I}{c} = \frac{\frac{1}{64}\pi\,d^4}{\frac{1}{2}d} = \frac{\pi\,d^3}{32}$. Hence, $\frac{W\,l}{8} = \frac{W \times (16 \times 12)}{8} = \frac{120{,}000 \times 3.1416 \times 4^3}{32}$, or $W = \frac{120{,}000 \times 3.1416 \times 64 \times 8}{16 \times 12 \times 32} = 31{,}416$ lb.

DEFLECTION OF BEAMS.

1404. The deflection, or amount of bending, produced in a beam by one or more loads is given by certain general formulas, whose derivation is too complicated to be given here. We shall give the formulas only, illustrating their application by examples.

In the third column of the Table of Bending Moments are given expressions for the value of the greatest deflection of a beam when loaded as shown in the first column. From this it is seen that the deflection s equals a constant (depending upon the manner of loading the beam and upon the condition of the ends—whether fixed or free), multiplied by $\frac{W\,l^3}{E\,I}$. Let a represent the constant and s the deflection; then,

$$s = a\frac{W\,l^3}{E\,I}. \qquad \textbf{(118.)}$$

In the above formula, E = coefficient of elasticity, and is to be taken from Table 25, Art. **1356**, l = length in inches, W = total weight supported in pounds, and I is the moment of inertia about the neutral axis; a has values varying from $\frac{1}{384}$ to $\frac{1}{3}$.

It will be noticed that the deflection is given for only nine cases; for any other manner of loading a beam than those here given, it is necessary to use the calculus to obtain the deflection.

EXAMPLE.—What will be the maximum deflection of a simple wooden beam 9 feet long, whose cross-section is an ellipse, having axes of 6 inches and 4 inches (short axis vertical), under a concentrated load of 1,000 pounds?

SOLUTION.—Use formula **118**, $s = a\frac{W l^3}{E I}$. From the Table of Bending Moments,

$$a = \frac{1}{48},\ W = 1{,}000 \text{ lb.},\ l = 9 \times 12 = 108 \text{ in.},$$

$$E = 1{,}500{,}000 \text{ and } I = \frac{\pi b d^3}{64} = \frac{\pi \times 6 \times 4^3}{64}.$$

Hence, $s = \frac{1{,}000 \times 108^3 \times 64}{48 \times 1{,}500{,}000 \times \pi \times 6 \times 64} = .9282''$.

1405. The principal use of the formula for deflection is to determine by its means the stiffness of a beam or shaft. In designing machinery, it frequently occurs that a piece may be strong enough to sustain the load with perfect safety, but the deflection may be more than circumstances will permit; in this case, the piece must be made larger than is really necessary for mere strength. An example of this occurs in the case of locomotive guides, and the upper guides of a steam engine when the engine runs under. It is obvious that they must be very stiff. In such cases it is usual to allow a certain deflection, and then proportion the piece so that the deflection shall not exceed the amount decided upon.

EXAMPLE.—The breadth of the guides of a certain locomotive is not to exceed $2\frac{1}{4}$ inches. Regarding the guides as fixed at both ends, (*a*) what must be their depth to resist a load of 10,000 pounds at the middle? The guides are made of cast iron and are 38 inches long between the points of support. (*b*) What weight would these guides be able to support with safety? The deflection must not exceed $\frac{1}{200}$ of an inch. The cross-section is, of course, rectangular.

SOLUTION.—Since the load comes on two guides, each piece must support $10,000 \div 2 = 5,000$ lb. In the formula,

$$s = \frac{1}{200} = a\frac{W l^3}{E I},\ a = \frac{1}{192} \text{ for this case, } W = 5,000,\ l = 38,$$

$$E = 15,000,000, \text{ and } I = \frac{b d^3}{12} = \frac{2\frac{1}{4} d^3}{12} = \frac{3}{16} d^3. \text{ Substituting,}$$

$$(a)\quad s = \frac{5,000 \times 38^3 \times 16}{192 \times 15,000,000 \times 3d^3} = \frac{1}{200},$$

$$\text{or } d = \sqrt[3]{\frac{5,000 \times 38^3 \times 16 \times 200}{192 \times 15,000,000 \times 3}} = 4.67'', \text{ nearly, say } 4\tfrac{11}{16}''. \quad \text{Ans.}$$

(*b*) To find the weight which these guides could support with safety, use formula **117**, $M = \frac{S_4 I}{f c}$, in which $M = \frac{W l}{8}$, $S_4 = 38,000$, $f = 10$,

$$\frac{I}{c} = \frac{b d^2}{6} = \frac{2\frac{1}{4} \times (4\frac{11}{16})^2}{6} = \frac{50,625}{6,144}. \quad \text{Substituting,}$$

$$W = \frac{38,000 \times 50,625 \times 8}{10 \times 6,144 \times 38} = 6,592 \text{ lb.} \quad \text{Ans.}$$

Hence, the beam is over 30% stronger than necessary, the extra depth being required for stiffness.

EXAMPLES FOR PRACTICE.

1. How much will a simple wooden beam 16 ft. long, 2 in. wide and 4 in. deep deflect under a load in the middle of 120 lb.? Ans. 1.106″.

2. What should be the size of a rectangular yellow pine girder 20 ft. long, to sustain a uniformly distributed load of 1,800 lb.? Assume a factor of safety for a varying stress. Ans. 5″ × 8″.

3. A hollow cylindrical beam, fixed at both ends, has diameters of 8 in. and 10 in. If the beam is 30 ft. long and is made of cast iron, (*a*) what steady load will it safely support at 15 ft. from one of the supports? (*b*) What force will be required to rupture the beam if applied at this point?

$$\text{Ans. } \begin{cases} (a)\ 8,158 \text{ lb.} \\ (b)\ 48,946 \text{ lb.} \end{cases}$$

4. A simple cylindrical wrought-iron beam, resting upon supports 24 ft. apart, sustains three concentrated loads of 350 lb. each, at distances from one of the supports of 5, 12, and 19 ft.; what should be the diameter of the beam to withstand shocks safely? Ans. 4.71″, say $4\frac{3}{4}''$.

5. Find the value of $\frac{I}{c}$ for a hollow rectangle whose outside dimensions are 10 in. and 13 in., and inside dimensions are 8 in. and 10 in.; (*a*) when the long side is vertical; (*b*) when the short side is vertical.

$$\text{Ans. } \begin{cases} (a)\ 179.103. \\ (b)\ 131\tfrac{1}{5}. \end{cases}$$

6. What is the deflection of a steel bar 1 in. square and 6 ft. long, which supports a load of 100 lb. at the center? Ans. .31104″.

7. Which will be the stronger, a beam whose cross-section is an equilateral triangle, one side measuring 15 in., or one whose cross-section is a square, one side measuring 9 in.? Both beams are of the same length. Ans. The one having the square cross-section.

8. A wooden beam of rectangular cross-section sustains a uniform load of 50 lb. per foot. If the beam is 8″ × 14″ and 16 ft. long, how much more will it deflect when the short side is vertical than when the long side is vertical? Ans. .055417″.

COMPARISON OF STRENGTH AND STIFFNESS OF BEAMS.

1406. Consider two rectangular beams, loaded in the same manner, having the same lengths and bending moments, but different breadths and depths. Then, $M = S\frac{I}{c} = S\frac{b\,d^2}{6}$ (1) and $M = S\frac{b_1 d_1^2}{6}$ (2). Dividing (1) by (2), $\frac{M}{M} = \frac{6\,S\,b\,d^2}{6\,S\,b_1\,d_1^2} = \frac{b\,d^2}{b_1\,d_1^2} = 1$, or $b\,d^2 = b_1\,d_1^2$ (3).

Equation 3 shows that, if both beams have the same depth, their strengths will vary directly as their breadths, i. e., if the breadths are increased 2, 3, 4, etc., times, their strengths will also be increased 2, 3, 4, etc., times. It also shows that, if the breadths are the same and the depths are increased, the strengths will vary as the square of the depth, i. e., if the depths are increased 2, 3, 4, etc., times, the strengths will be increased 4, 9, 16, etc., times. Hence, it is always best, when possible, to have the long side of a beam vertical. If the bending moments are the same, but the weights and lengths are different, $M = g\,W\,l$ (1) and $M = g\,W_1\,l_1$ (2), when g denotes the fraction $\frac{1}{2}$, $\frac{1}{4}$, etc., according to the manner in which the ends are secured, and the manner of loading. Dividing (1) by (2) $\frac{M}{M} = \frac{g\,W\,l}{g\,W_1\,l_1}$, or $W\,l = W_1\,l_1$ (3).

Equation 3 shows that if the load W or W_1 be increased, the length l or l_1 must be decreased; consequently, the strength of a beam loaded with a given weight varies inversely as its length, i. e., if the load be increased 2, 3, 4,

etc., times, the length must be shortened 2, 3, 4, etc., times, the breadth and depth remaining the same.

EXAMPLE.—If a simple beam, loaded in the middle, has its breadth and depth reduced one-half, what proportion of the original load could it carry?

SOLUTION.—In the preceding paragraphs, it was shown that the strength varied as the product of the breadth and the square of the depth, or $b_1 d_1^2 = \frac{1}{2} \times (\frac{1}{2})^2 = \frac{1}{8}$. Consequently, the beam can support only $\frac{1}{8}$ of the original load. Had the breadth remained the same, $(\frac{1}{2})^2 = \frac{1}{4}$ of the original load could have been supported. Had the depth remained the same, $\frac{1}{2}$ of the original load could have been supported.

EXAMPLE.—A beam 10 ft. long, loaded in the middle, has a breadth of 4 in. and a depth of 6 in. The length is increased to 12 ft., the breadth to 6 in., and the depth to 8 in.; how many times the original load can it now support?

SOLUTION.—The strength varies directly as the product of the breadth and square of the depth and inversely as the length, or as $\frac{b\,d^2}{l}$. If b, d, and l denote the original sizes, the strength of the two beams will be to each other as $\frac{b_1 d_1^2}{l} : \frac{b\,d^2}{l}$, or as $\frac{6 \times 8^2}{12} : \frac{4 \times 6^2}{10}$; $\frac{6 \times 8^2}{12} = 32$ and $\frac{4 \times 6^2}{10} = 14.4$. $\frac{32}{14.4} = 2\frac{2}{9}$. Consequently, the beam will support a load $2\frac{2}{9}$ times as great as the original beam.

1407. By a process of reasoning similar to that employed above, it can be shown that the maximum deflection of a beam varies inversely as the cube of the depth and directly as the cube of the length. In other words, if the depth be increased 2, 3, 4, etc., times, the deflection will be decreased 8, 27, 64, etc., times; and, if the length be increased 2, 3, 4, etc., times, the deflection will be increased 8, 27, 64, etc., times. Hence, if a beam is required to be very stiff, the length should be made as short and the depth as great as circumstances will permit.

COLUMNS.

1408. When a piece ten or more times as long as its least diameter or side (in general, its least transverse dimension) is subjected to compression, it is called a **column** or **pillar.**

The ordinary rules for compression do not apply to columns, for the reason that when a long piece is loaded beyond a certain amount, it buckles and tends to fail by flexure. This combination of flexure and compression causes the column to break under a load considerably less than that required to merely crush the material. It is likewise evident that the strength of a column is principally dependent upon its diameter, since that part having the least thickness is the part that buckles, or bends. A column free to turn in any direction, having a cross-section of $3'' \times 8''$, is not nearly so strong as one whose cross-section is $4'' \times 6''$. The strength of a very long column varies, practically, inversely as the square of the length; i. e., if a column *b* is twice as long as a column *a*, the strength of $b = (\frac{1}{2})^2 = \frac{1}{4}$ the strength of *a*, the cross-sections being equal.

1409. The conditions of the ends of a column play a very important part in determining their strength, and must always be taken into consideration. In Fig. 336, are shown three classes of columns. The column marked *a* is used in architecture, while the columns similar to *b* and *c* are used in bridge and machine construction.

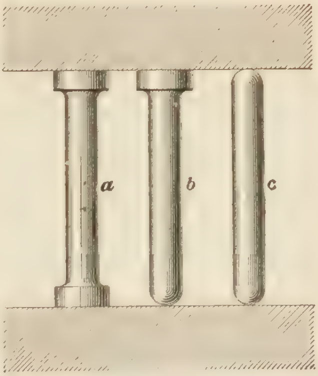


FIG. 336.

According to theory, which is confirmed by experiment, a column having one end flat and the other rounded, like *b*, is $2\frac{1}{4}$ times as strong as a column having both ends rounded, like *c*.

One having both ends flat, like *a*, is 4 times as strong as *c*, which has both ends rounded, the three columns being of the same length. If the length of *c* be taken as 1, the length of *b* may be $1\frac{1}{2}$, and that of *a* may be 2 for equal strength, the cross-sections all being the same; for, since the strengths vary inversely as the squares of the lengths, the strength of *c* is to that of *b* as $1 : \frac{1}{(1\frac{1}{2})^2}$ or as $1 : \frac{4}{9}$. But,

since b is $2\frac{1}{4} = \frac{9}{4}$ times as strong as c, $\frac{4}{9} \times \frac{9}{4} = 1$, or, the length of b being $1\frac{1}{2}$ times that of c, its strength is the same. Similarly, when a is twice as long as c its strength is the same.

1410. Columns like b and c do not actually occur in practice, an eye being formed at the end of the column and a pin inserted, forming what may be termed a *hinged end.* A steam engine connecting-rod is a good example of a column having two hinged ends, and a piston rod of a column having one end hinged and one end flat.

1411. There are numerous formulas for calculating the strength of columns, but the one that gives the most satisfactory results for columns of all lengths is the following:

$$W = \frac{S_2 A}{f\left(1 + \frac{A l^2}{g I}\right)}. \qquad \textbf{(119.)}$$

In this formula, W = load, S_2 = ultimate strength for compression, taken from Table 26, Art. **1357**, A = area of section of a column in square inches, f = factor of safety, l = length in inches, g = constant, to be taken from Table 30, and I = least moment of inertia of the cross-section—that is, the moment of inertia about an axis passing through the center of gravity of the cross-section and parallel to the longest side. In other words, if the column has a rectangular cross-section, whose longer side is b and shorter side d, the least moment of inertia is $\frac{b d^3}{12}$, the axis in this case being parallel to the long sides b. The values of g are given in the following table:

TABLE 30.

Material.	Both Ends Fixed.	One End Hinged.	Both Ends Hinged.
Timber	3,000	1,690	750
Cast Iron	5,000	2,810	1,250
Wrought Iron............	36,000	20,250	9,000
Steel	25,000	14,060	6,250

EXAMPLE.—A column whose cross-section is an I (see Table of Moments of Inertia) is used as a column having both ends fixed, $d = 8''$, $d_1 = 6''$, $b = 6''$, and $b_1 = 3\frac{1}{2}''$. If made of wrought iron, what steady load will it sustain with safety, the length being 10 ft.?

SOLUTION.—From the Table of Moments of Inertia, least I $= \frac{1}{12}(d\,b^3 - d_1\,b_1^3) = \frac{8 \times 6^3 - 6 \times 3.5^3}{12} = 122.5625$. $A = b\,d - b_1\,d_1 = 6 \times 8 - 3.5 \times 6 = 27$ sq. in. $S_2 = 55{,}000$, $f = 4$, $l = 10 \times 12 = 120''$ and $g = 36{,}000$. Therefore, $W = \dfrac{55{,}000 \times 27}{4\left(1 + \dfrac{27 \times 120^2}{36{,}000 \times 122.5625}\right)} = \dfrac{1{,}485{,}000}{4.35247} = 341{,}186$ lb., nearly.

Had the column been less than $10 \times 6 = 60$ in. $= 5$ ft. long, the safe load would have been $\dfrac{55{,}000 \times 27}{4} = 371{,}250$ lb. Had it been twice as long, it would have supported a safe load of only $\dfrac{55{,}000 \times 27}{4\left(1 + \dfrac{27 \times 240^2}{36{,}000 \times 122.5625}\right)} = 274{,}498$ lb.

1412. In the actual designing of a column, the size of the cross-section is not known, but the form (square, round, etc.) is known, also the length, material, condition of ends and load it is to carry. To find the size of the cross-section, substitute $\frac{S_2}{f}$ in formula **108**, for S, and solve for A, obtaining $A = \frac{Pf}{S_2}$. Substituting in this equation the values of $P\,(= W)$, f, and S_2, this gives the value of A for a short piece less than 10 times the length of the shortest side, or diameter. Assume a value of A somewhat larger than that just found, and dimension a cross-section of the form chosen so that its area shall equal that assumed. Calculate the moment of inertia and substitute the values of W, A, I, l, and g in formula **119**, and solve for $\frac{S_2}{f}$. If the result last found equals the value of $\frac{S_2}{f}$ taken from Tables 26 and 28, the assumed dimensions are correct; if larger, the assumed dimensions must be increased: if smaller, they should be

diminished, and in both cases the value of $\frac{S_2}{f}$ should be recalculated. An example will serve to illustrate the process.

EXAMPLE.—What should be the diameter of a steel piston rod 5 feet long, the diameter of the piston being 18 inches and the greatest pressure 130 pounds per square inch?

SOLUTION.—S_2 for this case $= 150{,}000$ lb. Since the piston rod is liable to shocks, a factor of safety of 10 should be used; hence, $\frac{S_2}{f} = \frac{150{,}000}{10} = 15{,}000$ lb. The load $W = 18^2 \times .7854 \times 130 = 33{,}081$ lb. $A = \frac{Pf}{S_2} = \frac{33{,}081}{15{,}000} = 2.2$ sq. in., nearly.

Assume that 3 sq. in. are needed. The diameter of a circle corresponding to an area of 3 sq. in. is $\sqrt{\frac{3}{.7854}} = 1.9544''$. Assume the diameter to be $1\frac{15}{16}'' = 1.9375$; the area will be $1.9375^2 \times .7854 = 2.9483$ sq. in. The value of $I = \frac{\pi d^4}{64} = \frac{3.1416 \times (1\frac{15}{16})^4}{64} = .69173$. $W = \frac{S_2 A}{f\left(1 + \frac{A l^2}{g I}\right)}$.

Consequently, $\frac{S_2}{f} = \frac{W}{A}\left(1 + \frac{A l^2}{g I}\right) = \frac{33{,}081}{2.9483}\left(1 + \frac{2.9483 \times (5 \times 12)^2}{14{,}060 \times .69173}\right) = 23{,}465$ lb.

As this value exceeds 15,000 lb., the diameter of the rod must be increased. Trying $2\frac{1}{8}''$ as the diameter, the area is 3.5466 sq. in., and $I = 1.00093$. Substituting these values as before, $\frac{S^2}{f} = \frac{33{,}081}{3.5466}\left(1 + \frac{3.5466 \times 60^2}{14{,}060 \times 1.00093}\right) = 17{,}790$ lb. This is still too large; hence, trying $2\frac{1}{4}''$, the area $= 3.976$ sq. in. $I = 1.258$ and

$$\frac{S_2}{f} = \frac{33{,}081}{3.976}\left(1 + \frac{3.976 \times 60^2}{14{,}060 \times 1.258}\right) = 15{,}052 \text{ lb.}$$

Consequently, the diameter should be $2\frac{1}{4}''$.

EXAMPLES FOR PRACTICE.

1. What safe steady load will a hollow cylindrical cast-iron column support, which is 14 feet long, outside diameter 10 inches, inside diameter 8 inches, and which has flat ends? Ans. 273,500 lb.

2. A hollow wooden column having a square cross-section is to support a steady load of 15,155 pounds. If the thickness of the side is $1\frac{1}{2}$ inches, length of column 20 feet, and the ends flat, what should be the length of the sides of the cross-section, outside and inside?

Ans. Outside, 9″; inside, 6″.

3. Suppose a wrought-iron connecting-rod to have a rectangular cross-section of uniform size throughout its length. If the diameter of the steam cylinder is 40 inches, steam pressure 110 pounds per square inch, and the length of the rod is $12\frac{1}{2}$ feet, what should be the dimensions of the cross-section of the rod? Ans. $5\frac{1}{2}'' \times 9''$.

1413. The preceding method for determining the dimensions of the cross-section, when the load and length are given, is perfectly general, and can, therefore, be used in every case. It is, however, somewhat long and cumbersome. For the special cases of square, circular, and rectangular columns, the following formulas may be applied, if preferred. They seem complicated, but, when substitutions are made for the quantities given, the formulas will be found of relatively easy application.

For square columns, the side c of the square is given by the formula

$$c = \sqrt{\frac{Wf}{2S_2} + \sqrt{\frac{Wf}{S_2}\left(\frac{Wf}{4S_2} + \frac{12l^2}{g}\right)}}. \qquad \textbf{(120.)}$$

For circular columns, the diameter d of the circle is given by the formula

$$d = 1.4142\sqrt{\frac{.3183\,Wf}{S_2} + \sqrt{\frac{.3183\,Wf}{S_2}\left(\frac{.3183\,Wf}{S_2} + \frac{16\,l^2}{g}\right)}}. \qquad \textbf{(121.)}$$

For rectangular columns, assume the shorter dimension (depth $= d$). Then the longer dimension (breadth $= b$) is given by the formula

$$b = \frac{Wf\left(1 + \frac{12\,l^2}{d^2 g}\right)}{d\,S_2}. \qquad \textbf{(122.)}$$

Should the dimensions given by the last formula be too much out of proportion, a new value may be assumed for d, and a new value found for b.

EXAMPLE.—Required the section of a square timber pillar to stand a steady load of 20 tons, the length of the column being 30 feet, and its ends both flat.

SOLUTION.—Here $S_2 = 8{,}000$ lb., $f = 8$, $g = 3{,}000$, $W = 40{,}000$ lb., $l = 30 \times 12 = 360$ in. These values, substituted in formula **120,** give

$$c = \sqrt{\frac{40{,}000 \times 8}{2 \times 8{,}000} + \sqrt{\frac{40{,}000 \times 8}{8{,}000}\left(\frac{40{,}000 \times 8}{4 \times 8{,}000} + \frac{12 \times 360^2}{3{,}000}\right)}}$$

$$= \sqrt{20 + \sqrt{40\left(10 + \frac{4 \times 36^2}{10}\right)}}$$

$$= \sqrt{20 + \sqrt{21{,}136}} = \sqrt{20 + 145.35}$$

$$= \sqrt{165.35} = 12.90 = 12\tfrac{7}{8}'', \text{ nearly, or say } 13''.$$

EXAMPLE.—Let it be required to solve the problem worked out by the general method in Art. **1412.**

SOLUTION.—Here $S_2 = 150{,}000$ lb., $f = 10$, $g = 14{,}060$, $W = 33{,}000$ lb., nearly, and $l = 5 \times 12 = 60$ in. From these data we have

$$\frac{.3183\ Wf}{S_2} = \frac{.3183 \times 33{,}000 \times 10}{150{,}000} = \frac{.3183 \times 11}{5} = .7003,$$

$$\frac{16\ l^2}{g} = \frac{16 \times 60^2}{14{,}060} = \frac{8 \times 360}{703} = \frac{4.0967}{4.7970}.$$

Then, formula **121,**

$$d = 1.4142\sqrt{.7003 + \sqrt{.7003 \times 4.80}}$$

$$= 1.4142\sqrt{.7003 + \sqrt{3.3614}} = 1.4142\sqrt{.7003 + 1.8334}$$

$$= 1.4142 \times 1.59 = 2.23 = 2\tfrac{1}{4}'', \text{ nearly,}$$

as found by the general or trial method.

The student may apply formula **122** to the solution of example 3 in the preceding article.

TORSION AND SHAFTS.

1414. When a force is applied to a beam in such a manner that it tends to twist it, the stress thus produced is termed **torsion.** In Fig. 337, $b\,c$ represents a beam fixed at one end; a load W is applied at the end of a lever arm $o\,n$, which twists the beam. If a straight line $c\,b$ is drawn parallel to the axis before the load is applied, it will be found, after the weight W has been hung from n, that the line $c\,b$ will take a position $c\,a$, forming a spiral. If the load

does not strain the material beyond its elastic limit, $c\ a$ will return to its original position $c\ b$ when W is removed. It will also be found that the angles $a\ c\ b$ and $a\ o\ b$ are directly proportional to the loads.

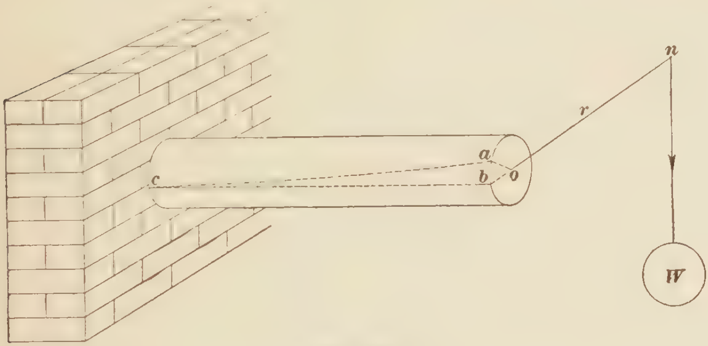


FIG. 337.

Torsion manifests itself in the case of rotating shafts. Instead of one end being fixed, as in the previous case, the reisistance which the shaft has to overcome takes the place of the force which before was necessary for fixing one end. Should the shaft be too small, the resistance will overcome the strength of the material and rupture it.

1415. The angle $a\ o\ b$, which may be called the angle of twist, plays an important part in the designing of shafts. For all solid shafts below 11 inches in diameter, the following formula may be used:

$$d = c\sqrt[4]{Pr} = c_1\sqrt[4]{\frac{H}{N}}, \qquad \textbf{(123.)}$$

in which $d =$ diameter of round shaft or the side of a square shaft in inches; $c =$ constant from Table 31; $P =$ force or weight applied to the end of the lever arm, in pounds; $r =$ length of lever arm in inches, from center of shaft to point of application of P; $c_1 =$ constant from same table; $H =$ horsepower transmitted, and $N =$ number of revolutions per minute.

TABLE 31.

Material.	c.		c_1.	
	Round.	Square.	Round.	Square.
Wrought Iron......	.31	.272	4.92	4.31
Cast Iron..........	.353	.309	5.59	4.89
Steel..............	.297	.26	4.7	4.11

EXAMPLE.—What should be the diameter of a wrought-iron crank shaft for a 16″ × 20″ steam engine, if the greatest steam pressure is to be 90 lb. per sq. in.? (Assume that the entire steam pressure is transmitted through the crank-pin at some point of the stroke).

SOLUTION.—Total pressure on piston $= 16^2 \times .7854 \times 90 = 18{,}095.616$, say 18,000 lb. $= P$ in formula **123.** $r = \frac{20}{2} = 10''$. Therefore,

$$d = c\sqrt[4]{Pr} = .31\sqrt[4]{18{,}000 \times 10} = 6.385''.$$

A $6\frac{3}{8}''$ shaft would be sufficiently large.

EXAMPLE.—What horsepower could be safely transmitted by a 7-inch cast iron square shaft making 80 revolutions per minute?

SOLUTION.—Formula **123** gives, $d = c_1\sqrt[4]{\frac{H}{N}}$, or $H = \frac{Nd^4}{c_1^4} = \frac{80 \times 7^4}{4.89^4}$ $= 335.93$, say 336 horsepower.

1416. If the diameter of a wrought-iron shaft is greater than 12.4 inches, of a cast-iron shaft greater than 10.3 inches, or of a steel shaft greater than 13.6 inches, the following formula should be used:

$$d = k\sqrt[3]{Pr} = k_1\sqrt[3]{\frac{H}{N}}, \qquad \textbf{(124.)}$$

k and k_1 being taken from Table 32. If the shaft is hollow (round), either of the two following formulas may be used:

$$P = q\left(\frac{d_1^4 - d_2^4}{d_1 r}\right), \qquad \textbf{(125.)}$$

or

$$H = q_1 N\left(\frac{d_1^4 - d_2^4}{d_1}\right), \qquad \textbf{(126.)}$$

d_1 and d_2 being the outside and inside diameters respectively and q and q_1 constants to be taken from Table 32.

TABLE 32.

Material.	k	k_1	q	q_1
Wrought Iron.......	.0909	3.62	1,335	.0212
Cast Iron...........	.1145	4.56	669	.0106
Steel...............	.0828	3.3	1,767	.028

EXAMPLE.—What horsepower can be safely transmitted by a hollow wrought-iron shaft making 60 revolutions per minute, and whose diameters are 9½ and 12 inches?

SOLUTION.—

$$H = q_1 N\left(\frac{d_1^4 - d_2^4}{d_1}\right) = .0212 \times 60\left(\frac{12^4 - 9.5^4}{12}\right) = 1,334.65 \text{ H.P.}$$

EXAMPLES FOR PRACTICE.

1. What should be the diameter of a steel shaft to transmit 500 horsepower at 200 revolutions per minute? Ans. 5.91″, say $5\frac{15}{16}$″.

2. How many horsepower will an 8″ round wrought-iron shaft transmit with safety, running at 150 R. P. M.? Ans. 1,048.5 H.P.

3. A hollow cast-iron shaft has an outside diameter of 10 inches and an inside diameter of 6 inches; at what speed should it be run to transmit 750 horsepower? Ans. 81.29 R. P. M.

4. A wrought-iron shaft 4 inches square runs at 110 revolutions per minute; what horsepower will it safely transmit? Ans. 81.6 H.P.

5. What should be the diameter of a wrought-iron shaft to transmit 6,000 horsepower at 100 revolutions per minute?

Ans. 14.172″, say $14\frac{3}{16}$″.

ROPES.

1417. The strength of hemp and manila ropes varies greatly, depending not so much upon the material and area of cross-section as upon the method of manufacture and the amount of twisting.

Hemp ropes are about 25% to 30% stronger than manila ropes or tarred hemp ropes. Ropes laid with tar wear better than those laid without tar, but their strength and flexibility are greatly reduced. For most purposes, the following formula may be used for the safe working load of any of the three ropes mentioned above:

$$P = 100\, C^2, \qquad \textbf{(127.)}$$

in which $P =$ working load in pounds and $C =$ circumference of rope in inches. This formula gives a factor of safety of from $7\frac{1}{2}$ for manila or tarred hemp rope to about 11 for best three strand hemp rope. When excessive wear is likely to occur, it is better to make the circumference of the rope considerably larger than that given by the formula.

1418. Wire rope is made by twisting a number of wires (usually 19) together into a strand and then twisting several strands (usually 7) together to form the rope. It is very much stronger than hemp rope, and may be much smaller in size to carry the same load.

For iron wire rope of 7 strands, 19 wires to the strand, the following formula may be used, the letters having the same meaning as in formula **127**:

$$P = 600\, C^2. \qquad \textbf{(128.)}$$

Steel wire ropes should be made of the best quality of steel wire; when so made they are superior to the best iron wire ropes. If made from an inferior quality of steel wire, the ropes are not as good as the better class of iron wire ropes. When substituting steel for iron ropes, the object in view should be to gain an increase of wear rather than to reduce the size. The following formula may be used in computing the size or working strength of the best steel wire rope, 7 strands, 19 wires to the strand:

$$P = 1{,}000\, C^2. \qquad \textbf{(129.)}$$

Formulas **128** and **129** are based on a factor of safety of 6.

1419. When using ropes for the purpose of raising loads to a considerable height, the weight of the rope itself must also be considered and added to the load. The weight of the rope per running foot, for different sizes, may be obtained from the manufacturer's catalogue.

EXAMPLE.—What should be the allowable working load of an iron wire rope whose circumference is $6\frac{3}{4}$ inches? Weight of rope not to be considered.

SOLUTION.—Using formula **128**,

$$P = 600 \times (6\tfrac{3}{4})^2 = 27{,}337.5 \text{ lb.}$$

EXAMPLE.—The working load, including weight, of a hemp rope is to be 900 pounds; what should be its circumference?

SOLUTION.—Using formula **127**,

$$C = \sqrt{\frac{P}{100}} = \sqrt{\frac{900}{100}} = 3''.$$

1420. In measuring ropes, the circumference is used instead of the diameter, because the ropes are not round and the circumference is not equal to 3.1416 times the diameter. For three strands the circumference is about 2.86 d, for seven strands about 3 d, d being the diameter.

CHAINS.

1421. The size of a chain is always specified by giving the diameter of the iron from which the link is made. The two kinds of chain most generally used are the **open link** chain and the **stud link** chain. The former is shown by (*a*), Fig. 338, and the latter by (*b*). The stud prevents the two sides of a link from coming together when under a heavy pull, and thus strengthens the chain.

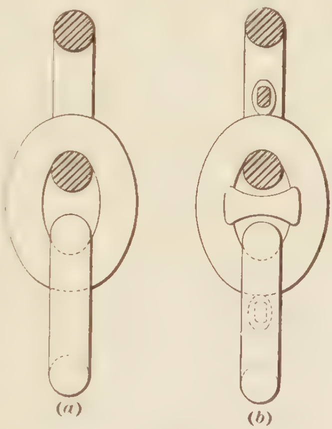


FIG. 338.

It is a good practice to anneal old chains which have become brittle by overstraining. This renders them less liable to snap from sudden jerks. The annealing process reduces their tensile strength, but increases their toughness and ductility, two qualities which are sometimes more important than mere strength.

Let P = safe load in pounds;

d = diameter of link in inches.

Then, for open link chains, made from a good quality of wrought iron,

$$P = 12{,}000\, d^2, \qquad \textbf{(130.)}$$

and, for stud link chains,

$$P = 18{,}000\, d^2. \qquad \textbf{(131.)}$$

Example.—What load will be safely sustained by a $\frac{3}{4}$-inch open link chain?

Solution.—Using formula **130,**

$$P = 12{,}000\, d^2 = 12{,}000 \times (\tfrac{3}{4})^2 = 6{,}750 \text{ lb.}$$

Example.—What must be the diameter of a stud link chain to carry a load of 28,125 pounds?

Solution.—Using formula **131,** $P = 18{,}000\, d^2$. Hence,

$$d = \sqrt{\frac{P}{18{,}000}} = \sqrt{\frac{28{,}125}{18{,}000}} = 1\tfrac{1}{4}''.$$

APPLIED MECHANICS.

1422. A **machine** is an assemblage of moving parts, together with a supporting frame so arranged as to utilize some external source of energy for the purpose of doing work.

In the operation of machinery, motion and force are communicated to one of the moving parts, and transmitted to the part where the work is done. During the transmission, both the motion and force are modified in direction and amount, so as to be rendered suitable for the purpose to which they are to be applied.

The moving parts are arranged to have certain definite motions relative to each other, the effect of which is to compel the piece where the work is done to have the required motion. The nature of these movements is independent of the amount of force transmitted; in other words, in a model of a machine, operated by hand, the relative motions of the parts will be precisely the same as in the machine itself, although, in the latter case, a great amount of power may be transmitted and much work done.

1423. **Kinematics** is that branch of Applied Mechanics which treats of the motions of the parts of a machine, without regard to the forces acting.

1424. The **dynamics of machinery** is that branch which treats of the forces acting in the operation of machinery. The dynamics of machinery depends upon the subject of Kinematics, since every change in force in a machine is the result of a change in motion. Thus, in the steam engine, if the motion of the piston be transmitted and modified so as to cause the periphery of the fly-wheel to move eight times as fast as the piston, the force exerted upon the belt would be only one-eighth of the steam pressure upon the piston.

In what follows, Kinematics will be treated of mainly, though in some cases the forces acting will be considered.

1425. Mechanism is a term applied to three or more parts of a machine, so combined that the motion of the first compels the motion of the other movable parts, according to the law depending upon the nature of the combination.

The terms *mechanical movement* and *mechanical motion* are often used as having the same meaning as *mechanism.* A machine is made up of a number of mechanisms.

1426. Driver and Follower.—That piece of a mechanism which causes motion is called the **driver,** and the one whose motion is effected is called the **follower.** In the case of belt or toothed gearing, the follower is often called the *driven* wheel.

1427. Right-handed rotation is rotation in the direction of the motion of the hands of a watch. **Left-handed rotation** is in the opposite direction Looking at a rotating pulley from one side, its rotation would be right-handed, if it turned in the same direction as the hands of a watch, held and looked at by the observer. Viewing the pulley from the other side, its rotation would be left-handed.

1428. Cycle of Motions.—When a mechanism is set in motion, and its parts go through a series of movements, which are repeated over and over in the same order, each series is called a **cycle of motions.**

1429. Velocity ratio is a term used to signify the comparative velocities of two pieces. Thus, if two gear wheels are so proportioned that one turns three times as fast as the other, their velocity ratio would be 3 or $\frac{1}{3}$, according as the more rapidly revolving gear was mentioned first or last.

LINK MECHANISMS.

1430. A bar, or other rigid body, connecting two elements or parts of a mechanism, is termed a **link.** In the steam engine, the crank and connecting-rod are links, and the engine frame may be considered to be a third or closing

link, one end of which supports the crank-shaft, the other end being slotted to guide the cross-head.

Links have special names, according to the machine or location in which they are used. Thus, a link which vibrates about a point is called a **beam, rocker,** or **lever,** and one which turns completely around a point is called a **crank.** A link connecting with an oscillating, or rotating, link is called by various names, as **connecting-rod, crank-rod, pitman, eccentric-rod, coupling-rod, parallel-rod,** etc. In practice, the word "link" is applied mainly to *slotted* links, such as the link in "Stephenson's link motion."

In what follows, when speaking of the length of a link, the distance between centers will be understood, or the distance A, in Fig. 339.

FIG. 339.

1431. The **center line of motion** of any mechanism is a straight line so drawn as to represent the general or mean direction in which one or more of the parts move. When the motion of the parts is not in a straight line, their deviation occurs equally on each side of this center line. Thus, in the steam engine, the center line of motion runs from the center of the cylinder to the center of the shaft, the connecting-rod vibrating equally on each side of the line.

1432. Levers.—Levers are used in mechanisms to guide a moving point, as the end of a moving rod, or to transfer motion from one line to another. There are three cases: (I) Levers whose lines of motion are parallel; (II) levers whose lines of motion intersect, and (III) levers having arms whose center lines do not lie in the same plane.

In proportioning levers, the following points should in general be observed. They apply to all three cases just mentioned:

(1.) When in mid position, the center lines of the arms should be perpendicular to the lines along which they give

or take their motions, so that the lever will vibrate equally each way.

(2.) If a vibrating link is connected to the lever, its point of attachment should be so located as to move equally on each side of the center line of motion of the link.

(3.) The lengths of the lever arms must be proportional to the distances through which they are to vibrate.

1433. Case I.—Reversing Levers.—An example of a lever which illustrates the foregoing principles is shown in Fig. 340. The crank $R\ S$ is the driver, and gives a

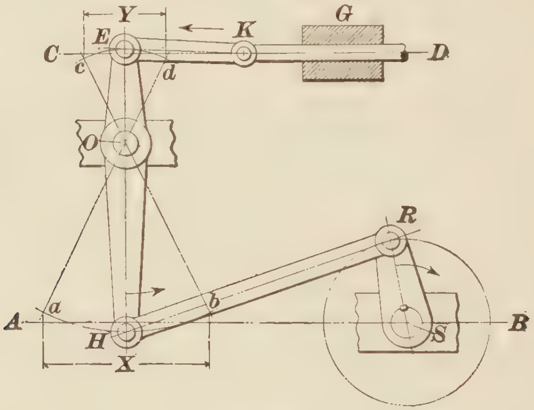


FIG. 340.

motion along the center line $A\ B$, which is transferred by the lever $E\ H$ to the line $C\ D$. The lever vibrates equally each way about its fulcrum or center O, as indicated by the lines $c\ b$ and $d\ a$. When in mid position, its center line $E\ H$ is perpendicular to the lines of motion $C\ D$ and $A\ B$. The horizontal distances traversed by points E and H, respectively, are proportional to the arms $E\ O$ and $H\ O$, or $Y : E\ O = X : H\ O$. The vibrating link $E\ K$ connects the point E with the rod $K\ D$, which is constrained to move in a straight line by the guide G, and in accordance with principle 2, the lever is so proportioned that point E will be as far above the center line of motion $C\ D$, when in mid position, as it will be below it in the two extreme positions;

that is, points c and d are as far below the line as point E is above it.

At the bottom of the lever, where the rod $H R$ connects with the crank, the same principle holds, point H being as far below the line $A B$ as points a and b are above it.

Frequently, the distance between the center lines $C D$ and $A B$ is given, and the extent of the motion along these lines, from which to proportion the lever. A correct solution to this problem is troublesome by calculation, because it is not known at the start how far above and below their respective lines of motion the points E and H should be.

1434. It may easily be done graphically, however, as shown in Fig. 341. Draw the center lines of motion $C D$ and $A B$ and a center line $S T$ perpendicular to them. Draw $M E$ parallel to $S T$ at a distance from it equal to $\frac{1}{2} Y$, or half the stroke along $C D$; also, the parallel line $H N$, on the other side of $S T$, and at a distance from it equal to $\frac{1}{2} X$, or half the stroke along $A B$. Connect points M and N by a straight line; where this line intersects $S T$, as at O, will be the center or fulcrum of the lever. With O as a center, find by trial the radius of an arc which will cut $S T$ as far below the line $A B$ as it does $H N$ above this line, or so that the distance m will equal the distance n. As an aid in determining the correct radius, describe an arc cutting $S T$, with O as a center and a radius $O N$. The

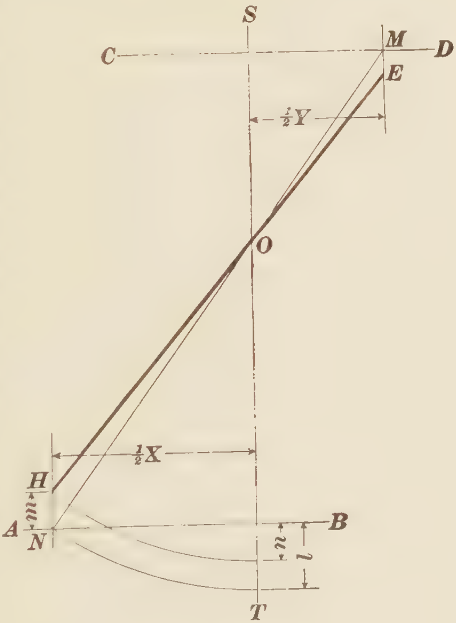


FIG. 341.

distance n will be a little more than $\frac{1}{2}$ l. Now, draw a straight line through points H and O. The part included between $H N$ and $M E$ determines the length of the lever. In this case, the length of the shorter arm is equal to $O E$, and of the longer arm, $O H$.

1435. Non-Reversing Levers.—Fig. 342 shows the same construction applied to a lever in which the center O is at one end of the lever. This lever does not reverse the motion like the previous one, since when the motion along $A B$ is to the right or left, the motion along $C D$ will be in the same direction. The figure is lettered like the preceding one, so that the construction will be easily understood.

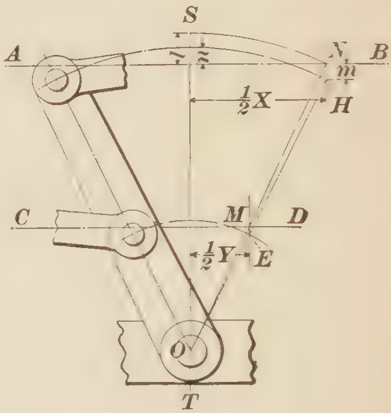


FIG. 342.

1436. It is often desirable that a lever mechanism shall reproduce upon a greater or smaller scale, along one line, the exact motion that occurs along another line; that is, that every change in the rate of the motion along one line shall reproduce a corresponding change along the other line. Figs. 343 and 344 illustrate three indicator-reducing motions that accomplish this.

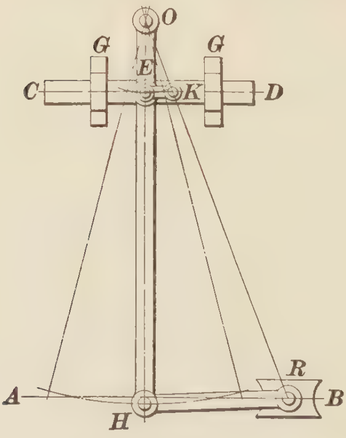


FIG. 343.

In Fig. 343 the lower end of the lever attaches to the cross-head of the engine through the swinging link $H R$. The indicator string is fastened to the bar $C D$, which receives its motion from the lever through

the link $E K$, and slides through the guides G, G, in a direction parallel to the line of motion A B of the cross-head. In order that the bar $C D$ shall have the same kind of motion

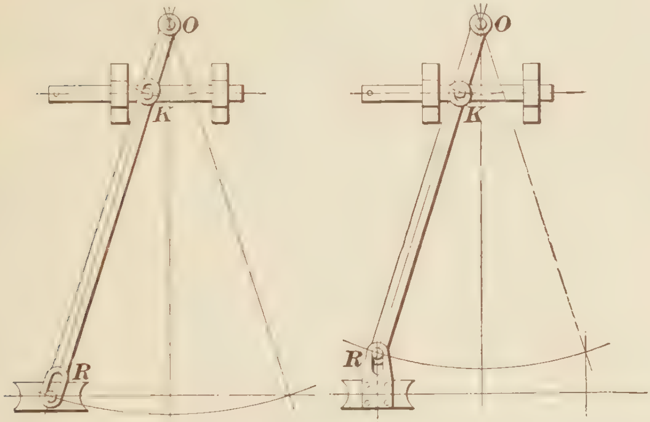


FIG. 344.

as the cross-head, it is only necessary that the links E K and H R shall be proportional to their respective lever arms; thus $O H : H R = O E : E K$. The pins must be so placed that the connecting links will be parallel; if parallel at one point of the stroke, they will be so at all points.

1437. It is to be observed that the pins O, K, and R are in one straight line, and, in general, it may be said that any arrangement of the lever which will keep these three pins in a straight line for all points of the stroke will be a correct one. In Fig. 344 are two such arrangements. In the first, the pins K and R are fast to the slide and cross-head, respectively, and slide in slots in the lever. In the second, they are fast to the lever, the slots being in the cross-head and slide. In both, the pins K and R are in a straight line with the pin O during the whole stroke.

1438. Case II.—Bell-Crank Levers.—Levers whose lines of motion intersect are termed **bell-crank** levers.

In Fig. 345, suppose the angle $C\,A\,B$, made by the lines

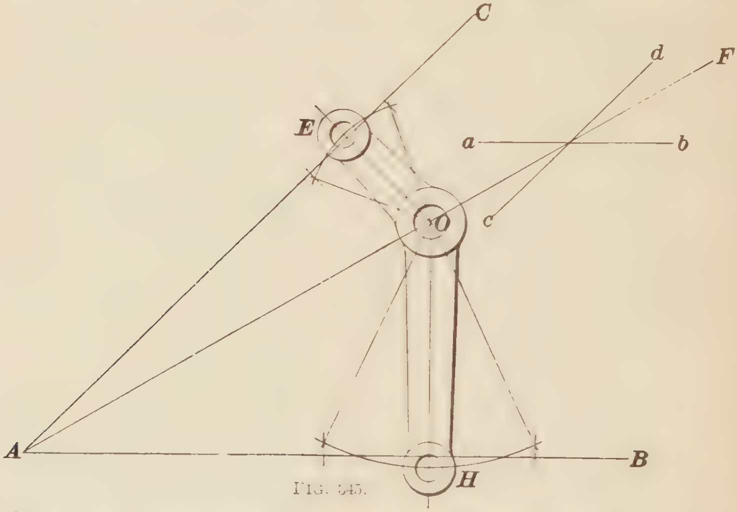


FIG. 345.

of motion, to be given, and that the motion along $A\,B$ is to be twice that along $A\,C$. Draw $c\,d$ parallel to $A\,C$ at any

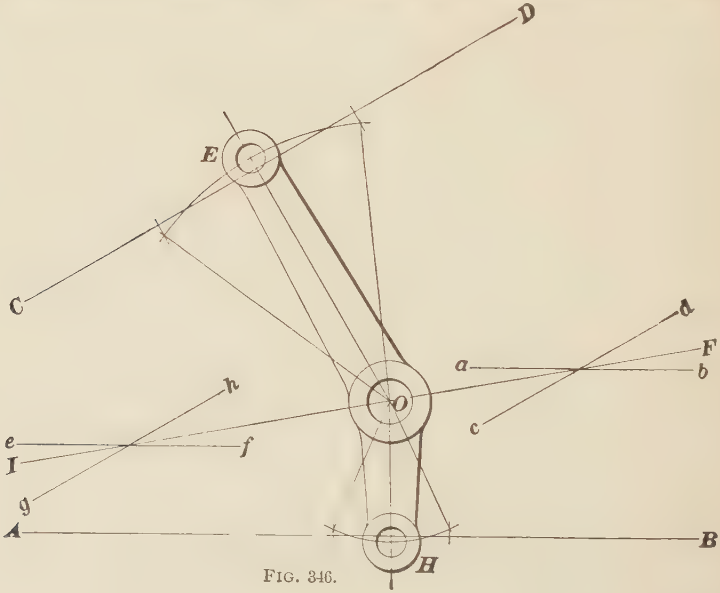


FIG. 346.

convenient distance from *A C*. Draw *a b* parallel to *A B* and at a distance from *A B* equal to twice the distance of *c d* from *A C*. Through the intersection of these two lines and the apex *A* of the angle, draw the line *A F*. Then the center *O* of the bell-crank may be taken at any point on *A F* suited to the design of the machine. Having chosen point *O*, draw the perpendiculars *O E* and *O H*, which will be the center lines of the lever arms.

1439. In Fig. 346 a construction is shown that may be employed when the two lines *C D* and *A B* do not intersect

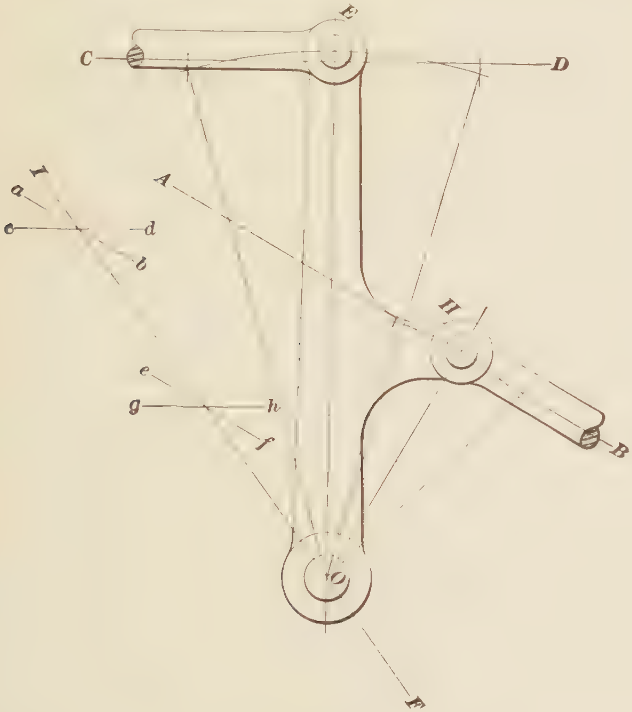


FIG. 347.

within the limits of the drawing. In Fig. 347 the same construction is applied to a non-reversing lever, in which the center *O* falls outside of the lines *A B* and *C D*. The figures

are lettered alike, and the following explanation applies to either: Draw $c\ d$ parallel to $C\ D$, and $a\ b$ parallel to $A\ B$, as before, so that the distance of $c\ d$ from $C\ D$: distance of $a\ b$ from $A\ B$ = amount of motion along $C\ D$: amount of motion along $A\ B$. Again, draw lines $g\ h$ and $e\ f$ in exactly the same way, but taking care to get their distances from $C\ D$ and $A\ B$ different from those of the lines just drawn. Thus, if $c\ d$ should be six inches from $C\ D$, make $g\ h$ some other distance, as four inches, or eight inches, and then draw $e\ f$ at a proportionate distance from $A\ B$. Through the intersections of $a\ b$ with $c\ d$, and of $e\ f$ with $g\ h$, draw the line $I\ F$, which will be the line of centers for the fulcrum O.

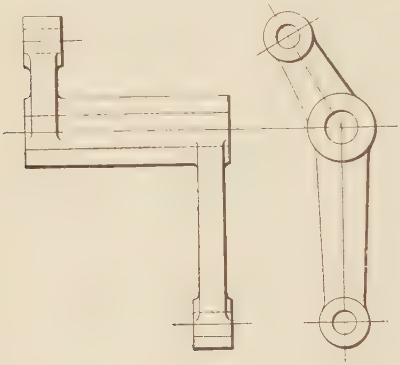


FIG. 348.

1440. Case III.—Levers falling under this case are usually bell-crank levers, with their arms separated by a long hub, so as to lie in different planes. They introduce no new principle. See Fig. 348.

1441. Crank and Connecting-Rod.—Fig. 349 is a diagram of the crank mechanism used in steam engines,

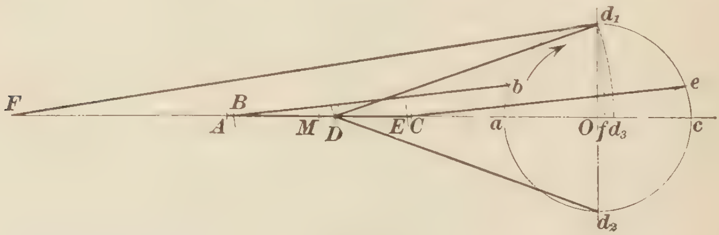


FIG. 349.

power pumps, etc. $A\ C$ represents the stroke of the crosshead, O the center of the crank-shaft, and $a\ d_1\ c\ d_2$ the path described by the crank-pin, called the crank-pin circle.

In the crank motion, the relative positions of the cross-head and crank-pin vary at every point of the stroke. This irregularity, which has an important influence in the design of steam-engine valve gears, may readily be observed by plotting a few points of the motion. Having drawn the crank-pin circle and the center line of motion $A\ c$, set the compasses to a radius equal to the length of the connecting-rod, which, in this case, is three times the length of the crank, or $3 \times O\ a$. With a as a center, strike an arc at A, and with c as a center, an arc at C. $A\ C$ equals the length of the stroke of the cross-head, A and C being the extreme positions of the stroke. The corresponding positions a and c of the crank-pin are known as the **dead points** or **dead centers**, because the crank can not be started at these points by a direct pressure on the connecting-rod.

With some point near a on the crank-pin circle, as b, for a center, and with the same radius as before, strike arc B on the stroke line. Mark point e on the crank-pin circle, so that arc $e\ c = b\ a$; with e as a center, and with the same radius, strike arc E. It will be seen that the distance $E\ C$ is less than $A\ B$, which shows that the relative motion during the first half stroke is different from that during the second half. The reason for this is that one-half the crank-pin circle curves *towards* the cross-head, and the other half *away* from it.

The greatest irregularity occurs when the crank is in its middle positions, or at points d_1 and d_2. With either point as a center, strike an arc, as before, which will fall at D, a distance equal to $M\ D$ from the mid-stroke position of the cross-head. Suppose, the crank to turn in the direction of the arrow, the cross-head will have *passed* mid position M, when the crank-pin reaches its middle point at d_1; on the return stroke the reverse will be true, the crank-pin reaching point d_2 *before* the cross-head reaches M. Hence, during the forward stroke, the cross-head *moves ahead* of the crank; on the return stroke, the cross-head *lags behind* the crank.

The common way of plotting the motion is as follows: Take for example, the point d_1. With d_1 as a center, strike the arc at D, as before; with D as a center and same radius, describe arc $d_1\ d_3$. $O\ d_3$ is the displacement of the cross-head from mid stroke. Point f was obtained in the same way by increasing the length of the connecting-rod to $F\ d_1$, *showing that the longer the connecting-rod, the less the irregularity.*

1442. Crank and Slotted Cross-Head.—If the connecting-rod in Fig. 349 be increased to a very great length, an arc drawn through d_1, corresponding to the arcs $d_1\ d_3$, and $d_1\ f$, would be nearly a straight line coinciding with $d_1\ O$, and the horizontal movement of the crank-pin would, therefore, be practically the same as that of the cross-head. If the connecting-rod were increased to an *infinite* length, the two movements would be exactly the same. Fig. 350 shows the crank and slotted cross-head mechanism by which this

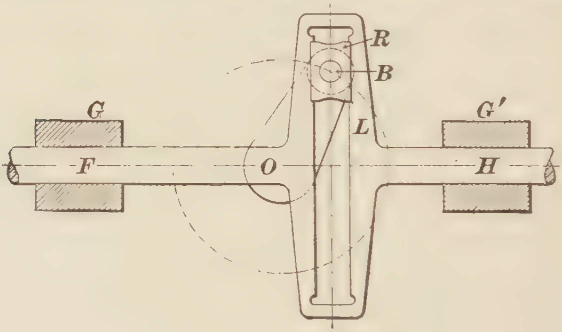


FIG. 350.

is accomplished. Consider the crank $O\ B$ as the driver. The crank-pin B is a working fit in the block R which is arranged to slide in the slotted link L. The rods F and H are rigidly attached to the link, and are compelled to move in a straight line by the guides G, G'. As the crank revolves, the rods F and H are given a horizontal motion exactly equal to the horizontal motion of the pin B.

This mechanism is often applied to steam pumps, where one of the rods, as F, is the steam piston rod, and the other

is the plunger rod. A fly-wheel is driven by means of the slotted link and crank, and its kinetic energy makes it possible to cut off the steam before the end of the stroke. Without the fly-wheel full steam pressure must be carried throughout the stroke.

1443. In Fig. 350, if the crank rotates uniformly, the motion of the sliding rods is said to be **harmonic,** and the mechanism itself is often called the **harmonic-motion** mechanism. Harmonic motion may be defined as the motion executed by the foot of a perpendicular let fall on the diameter of a circle from a point moving with uniform velocity along the circumference.

1444. Slow-Motion Mechanism.—A mechanism consisting of two connected levers, or of a crank and lever, can be proportioned to produce a slow motion of one of the levers.

Such a combination is shown in Fig. 351, where two levers, A and B, are arranged to turn on fixed centers, and are connected by the rod R. Lever A is actuated by the handle H, secured to the same shaft. If H and lever A be turned left-handed, lever B will turn right-handed, but with a decreasing velocity, which will become zero when the lever A reaches position A_1 in line with the rod, which will then be in position R_1. Any further motion of A will cause B to return towards its first position, its motion being slow at first and then faster. The mechanism, it will be observed, is proportioned contrary to the principles stated in Art. **1432,** and produces a motion that is variable, but very powerful.

To obtain the greatest advantage, the lever B should be so placed that it will occupy a position perpendicular to the link R at the instant when A and R are in line. To lay out the motion, therefore, supposing the positions of the centers and lengths of the levers to be known, describe arc $b\ a$ about the center O, with a radius equal to the length of lever B. Through C, the center of lever A, draw the line $M\ N$ tangent to the arc just drawn. R and A must then be in line

along the line $M N$, and lever B must be perpendicular to it when in position B_1. Generally, there will be a certain required amount of movement for lever B. To secure this, draw B in its extreme left-hand position; then with C as a center, and a radius equal to the length of A, strike the arc

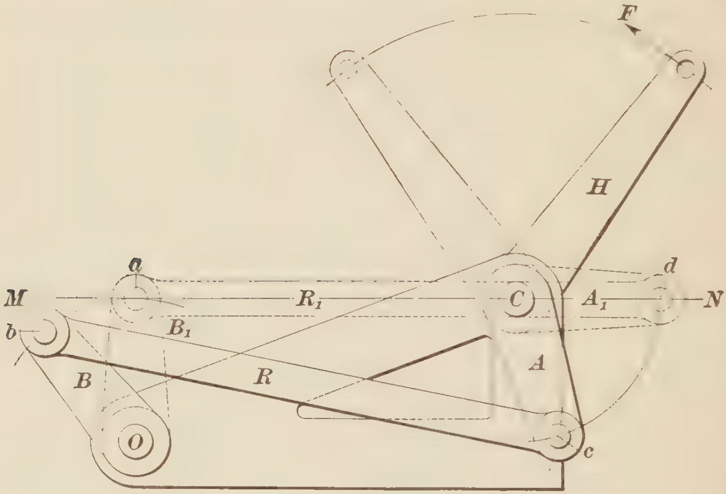


FIG. 351.

$d c$. Set the compasses to length $a d$, and from b as a center strike an arc cutting the arc $c d$. $C c$ will be the second position of lever A.

In the above combination, if the parts were proportioned to allow A to rotate like a crank, R and A would come into line twice during each revolution.

This mechanism has been used to operate platen printing presses, where oscillation of the handle H moves the platen to and from the type, through the lever B. It is also used to operate the exhaust valves in Corliss engines, in a manner which will be explained later.

1445. The **toggle-joint,** shown in Fig. 352, is a modification of the foregoing mechanism, and is much used in presses, punching and shearing machinery, etc. It consists of the two links $O R$ and $C R$, point O being fixed and point

C constrained to move in a horizontal line. A considerable movement of point R produces a very small movement in C,

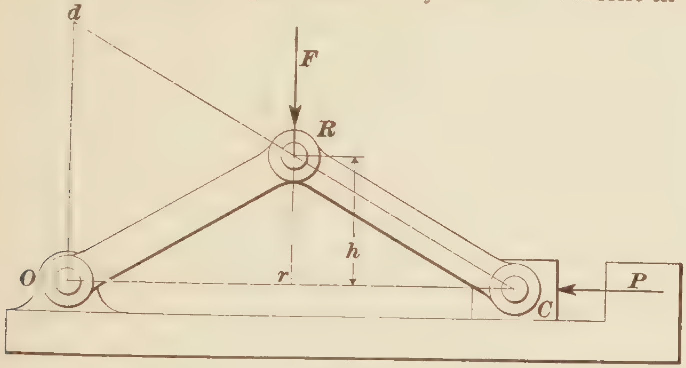


FIG. 352.

and consequently a pressure F, applied at R, can be made to produce a powerful pressure, or thrust, in a direction opposite to that of the arrow P.

THE FORCES ACTING IN LINK MECHANISMS.

1446. In order to understand how the forces act in a link mechanism, it will be necessary to again refer to the subject of moments, which is treated of in Elementary Mechanics.

The **moment of a force** *about any point is the product of the magnitude of the force and the perpendicular distance from the point to the line of action of the force.*

The tendency of a force to rotate a body about a point is measured by the *moment* of the force about that point. For example, let the crank C, in Fig. 353, be pivoted at O, and suppose a force P to act upon the crank-pin in the direction shown. Then, if the perpendicular distance from O to the line of action of the force be $O\,A$, the *moment of the force* tending to rotate the crank about O is $P \times O\,A$. If the force be stated in pounds and the perpendicular distance in inches, the product will

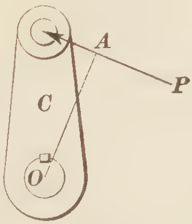


FIG. 353.

be in *inch-pounds;* if the perpendicular distance be given in

feet, the product will be in foot-pounds. These expressions, however, bear no relation to *foot-pounds of work*, and the student must avoid confusing the two.

In the case of a shaft having a pulley, gear, crank, or lever attached to it upon which a force acts tending to cause rotation of the shaft, the moment of the force is generally called the **twisting moment.**

1447. If two or more forces in one plane act upon a body, and are in equilibrium, then *the sum of the moments which tend to turn the body in one direction about a point is equal to the sum of the moments of the forces which tend to turn the body in the opposite direction about the same point.* Or, to state the principle more concisely, *the opposing moments about the point are equal.* This is called the **principle of moments.**

In the crank and connecting-rod mechanism, shown in outline in Fig. 354, the tendency of the force P to cause

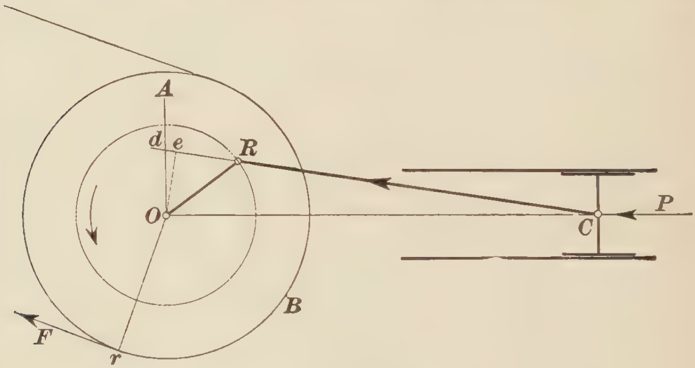


FIG. 354.

rotation of the crank about point O may be determined by resolving this force into two forces, one of which acts along the connecting-rod in the direction of the arrow, and which we will call P', and the other of which acts in a direction perpendicular to the guides. Draw $O\ e$ perpendicular to $C\ R$ produced; then, assuming the parts to be stationary for the instant, so that the effects of inertia may be neglected,

the twisting moment $= P' \times O\,c$. A simpler method, however, is the following: Let the force P act in the direction of the line $O\,C$. From O draw $O\,A$ perpendicular to $O\,C$, and note the point d where it intersects the center line of the connecting-rod, or of the center line produced. Then, it can be proved that the twisting moment about O due to the force P is $P \times O\,d$.

If a belt pulley B is attached to the shaft, the force P will be resisted by the pull F of the belt; and, by the *principle of moments*, $F \times O\,r = P \times O\,d$; whence,

$$F = \frac{P \times O\,d}{O\,r}. \qquad \textbf{(132.)}$$

EXAMPLE.—In a power pump, if a belt pull of 120 pounds is exerted upon the rim of the driving pulley, 24 inches in diameter, and the crank is in such a position that the distance $O\,d$ (Fig. 354) = 2 inches, what is the water pressure per square inch upon the pump plunger, if its diameter is 4 inches?

SOLUTION.—Radius of pulley = 12″ = $O\,r$. From formula **132**, $120 = \frac{P \times 2}{12}$, whence $P = 720$ lb. Area of piston = 12.57 sq. in. $720 \div 12.57 = 57.28$ lb. per sq. in. Ans.

1448. A drawing in which the arms, rods, and links of a mechanism are indicated by their center lines only is called **a skeleton diagram.**

Fig. 355 is a skeleton diagram of the motion shown in Fig. 351, the different parts being in the same position. Draw the line $C\,s$ perpendicular to $b\,c$, and suppose a force F to act at the end of the lever $C\,h$. By the principle of moments, the pull along $b\,c$ would then be such that $F \times C\,h$ = pull on $b\,c \times C\,s$, or

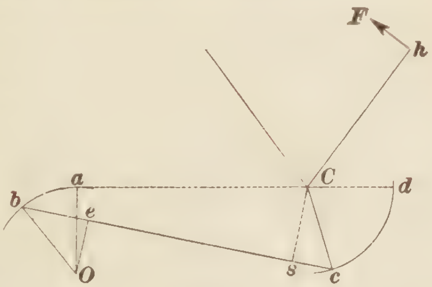


FIG. 355.

$$\text{Pull on } b\,c = \frac{F \times C\,h}{C\,s}. \qquad \textbf{(133.)}$$

The force necessary to balance the pull on $b\ c$ would be supplied by the resistance to motion by whatever force might be applied to the shaft O or the lever $b\ c$, and would be felt at the point b. If it were desired to find the twisting moment about the point O, we would multiply the pull on $b\ c$ by the perpendicular distance $O\ c$.

EXAMPLE.—In Fig. 355, if $F = 25$ lb., $C\ h = 2$ ft., $C\ s = \frac{1}{10}''$, and $O\ c = 3''$, (*a*) what would be the pull at b? (*b*) What is the twisting moment about O?

SOLUTION.—(*a*) Pull $= \dfrac{25 \times 24}{\frac{1}{10}} = 25 \times 24 \times 10 = 6{,}000$ lb. Ans.

(*b*) Twisting moment $= 6{,}000 \times 3 = 18{,}000$ inch-pounds. Ans.

1449. We will now consider the forces acting in the toggle-joint. In Fig. 356, let a pressure F be exerted upon

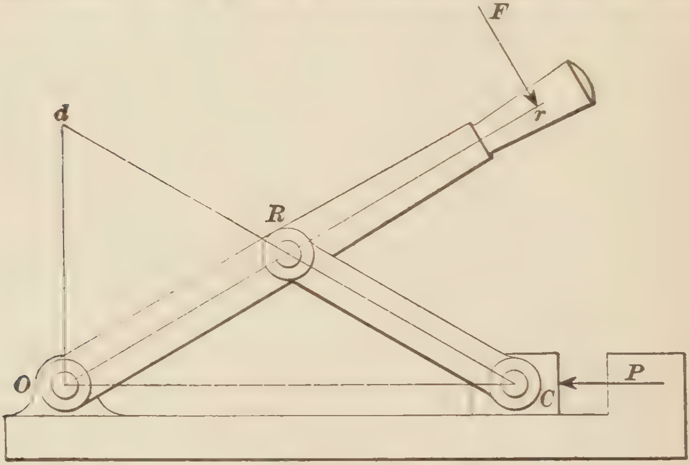


FIG. 356.

the handle in a direction at right angles to $O\ r$. The twisting moment about O is $F \times O\ r$, and the case becomes exactly similar to that of the connecting-rod crank, $O\ R$ corresponding to the crank and C to the cross-head. From O erect the perpendicular which intersects the center line of the link $C\ R$, extended, at d. Then, as before, $P \times O\ d = F \times O\ r$. It will here be more convenient to have the formula in terms of P. Hence,

$$P = \frac{F \times O\ r}{O\ d}. \qquad \textbf{(134.)}$$

The same formula applies when the force F acts as in Fig. 352. Drawing the line $O\ r$ perpendicular to the line of action of F, we have the twisting moment $F \times O\ r$ resisted by $P \times O\ d$.

1450. When the two links $O\ R$ and $R\ C$ are equal in length, the height h of the point R above a straight line drawn through points O and C will equal $\frac{1}{2}\ O\ d$. Hence, for equal links, formula **134** may be written,

$$P = \frac{F \times O\ r}{2\ h}. \qquad \textbf{(135.)}$$

EXAMPLE.—In Fig. 18, if $O\ r = 30''$, $O\ d = 6''$, and $F = 100$ lb., what thrust would be produced by the block C?

SOLUTION.—$P = \frac{100 \times 30}{6} = 500$ lb. Ans.

EXAMPLES FOR PRACTICE.

1. In Fig. 355, let $F = 250$ lb., $C\ h = 2$ ft., $C\ s = 2$ in., and $b\ O = 10$ in. If, in this position, the lever $b\ O$ is perpendicular to $b\ c$, what force would have to be exerted at a point midway between b and O, in order to resist the force F? Ans. 6,000 lb.

2. If the piston of an engine is 6 inches in diameter and the steam pressure 45 lb. per sq. in., what would be the tangential pressure at point R, in Fig. 354, when the crank is in such a position that $O\ d =$ 3 in., the length of the crank being 8 in.? Ans. 477.13 lb.

3. What thrust would be exerted by the block C, in Fig. 356, if the force F were to act in a vertical instead of a slanting direction? Take $F = 100$ lb., $O\ r = 60$ in., $O\ d = 1$ in., and distance of point r above the line $O\ C = 8$ in. Ans. 5,946½ lb., nearly.

QUICK-RETURN MOTIONS.

1451. Quick-return motions are used in shapers, slotters and other machines, where all the useful work is done during the stroke of a reciprocating piece in one direction. During the working stroke the tool must move at a suitable cutting speed, while on the return stroke, when no work is performed, it is desirable that it should travel as rapidly as possible.

Vibrating-Link Motion.—The mechanism shown in Fig. 357 has been applied to shaping machines operating on

metal. Motion is received from the pinion P, which drives the gear G. The pin b is fast to the gear, and pivoted to it is the block B, which is fitted to the slot of the link $C\,D$. As the gear rotates, the pin describes the circle $b\,e\,d\,c$, the block sliding in the slot of the link $C\,D$, causing $C\,D$ to oscil-

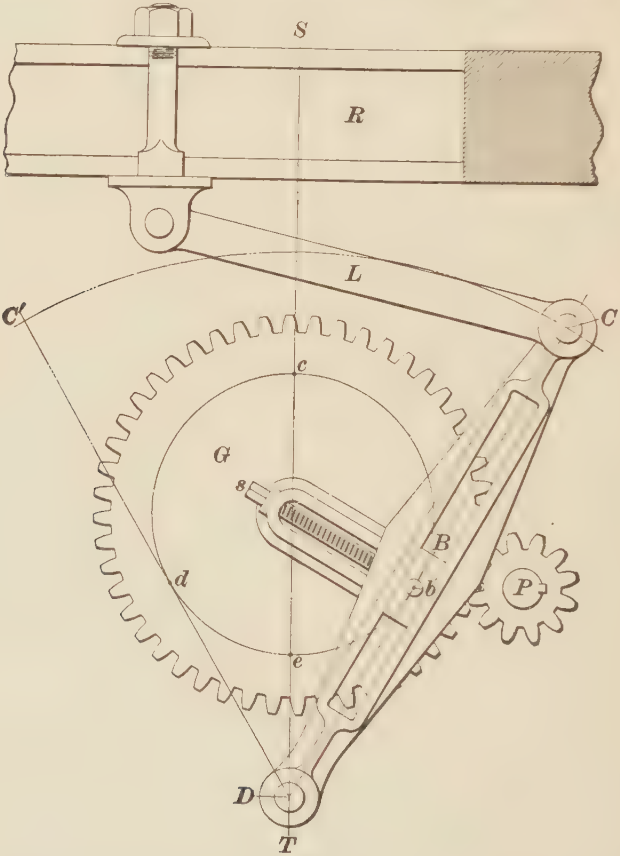


FIG. 357.

late about the point D, as indicated by the line $C\,C'$ in the figure. The rod L connects the upper end of the link with the tool slide, or "ram," R, which, therefore, oscillates with it, but is constrained by guides (not shown) to move in a straight, horizontal line.

During the cutting stroke, the pin b travels over the arc $d\ c\ b$, or around the greater arc included between the points of tangency of the center lines $C'\ D$ and $C\ D$. During the return stroke the pin passes over the shorter arc $b\ e\ d$, and as the wheel G revolves with a uniform velocity, the lines of the forward and return strokes will be to each other as the length of the arc $d\ c\ b$ is to the length of the arc $b\ e\ d$. The throw of the slotted link and the travel of the tool can be varied by the screw s, which moves the block B to and from the center of the gear. The rod L, instead of vibrating equally above and below a center line of motion, is so arranged that the force moving the ram during the cutting stroke will always be downwards, causing it to rest firmly on the guide.

1452. To lay out the motion, proceed as follows: Draw the center line $S\ T$, Fig. 358, and parallel to it the line $m\ n$, the distance between the two being equal to one-half the longest stroke of the tool. About O, which is assumed to be the center of the gear, describe the circle $b\ d\ c$ with a radius equal to the distance from the center of G to b (Fig. 357) when set for the longest stroke. Divide the circumference of the circle into an upper and lower arc, extending equally on each side of the center line, and having the same ratio as the forward and backward strokes. In this case the return is 2 to 1, and the circle is divided into three equal parts, as shown at b, d, and c, thus making the arc $d\ b$ equal to one-half $d\ c\ b$. Draw the radial lines $O\ b$ and $O\ d$. Through b draw $C\ D$, perpendicular to $O\ b$; the point D, where it intersects $S\ T$, will be the fulcrum, and the point C, where it intersects $m\ n$, the upper end of the slotted lever. Through C draw the horizontal line $C\ C'$, making $C'\ E$ equal to $C\ E$. Draw $C'\ D$, which should be tangent to the circle at d, thus giving the other extreme position of the lever. It is to be observed that for a quick return of 2 to 1, the only condition is that lines $C\ D$ and $C'\ D$ shall be perpendicular to $O\ b$ and $O\ d$, respectively; points C and D and the length of radius $O\ b$ can be varied considerably.

1453. To plot the motion, draw the center line of motion $R\ L$ through the point at which the connecting-rod attaches to the tool slide. Divide the circle $d\ c\ b$ into a number of equal parts, as *1*, *2*, *3*, etc., and from D draw lines through these points, extending to the arc $C'\ C$. Number the points of division on the circle, and give corresponding numbers to the points of intersection on the arc $C'\ C$. With these last points as centers, and with a radius equal to the length of the connecting-rod L, in Fig. 357, strike arcs

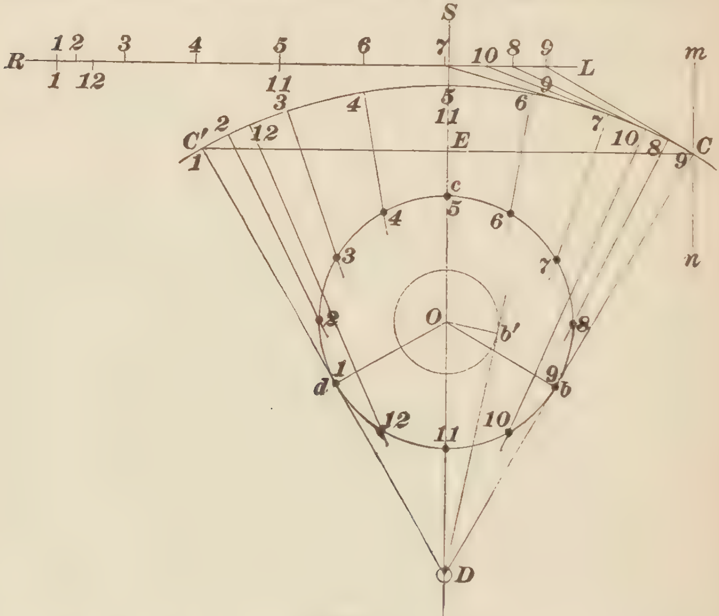


FIG. 358.

cutting the line $R\ L$, and number these intersections so that they will correspond to the other points. In Fig. 358, supposing the gear to turn with a uniform motion, the tool slide will move along $R\ L$ from point *1* to *2* during the first $\frac{1}{12}$ of a revolution; during the next $\frac{1}{12}$ revolution, from point *2* to *3*, etc., on the forward stroke. On the return stroke, from point *9* to *10*, *11*, *12*, and *1*, the motion is much less uniform.

Another point about this motion is that, as the radius $O\ b$ is diminished to shorten the stroke, the return becomes less

rapid, as can be seen from the dotted lines in the figure, which show the radius $O\ b$ shortened to $O\ b'$, and the corresponding position of $C\ D$.

1454. Whitworth Quick-Return Motion.—This mechanism is shown in principle in Fig. 359. The pin b, inserted in the side of the gear G, gives motion to the slotted link $C\ D$, as in the vibrating link motion. This motion closely resembles the previous one, the difference being that the center D, of the slotted link, lies *within* the circle described by the pin b, while in the previous case it lies *without* it. To accomplish this result, a pin P is provided for the gear to turn upon, and is made large enough to include another pin D, placed eccentrically within it, which acts as the center for $C\ D$. With this arrangement, the slotted link follows the crank-pin during the complete revolution, instead of

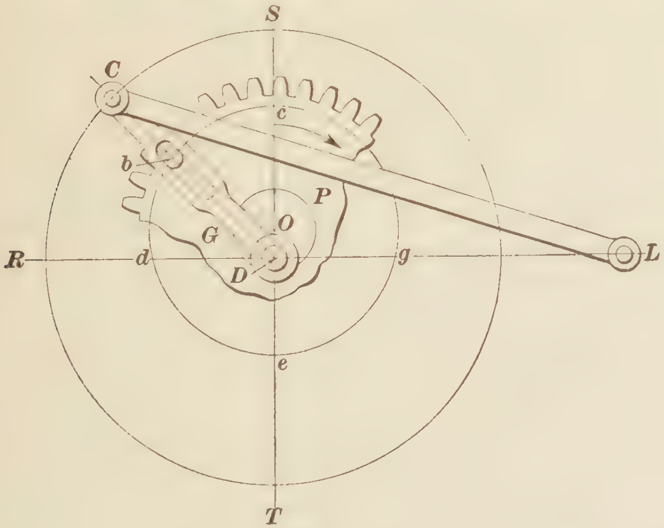


FIG. 359.

vibrating, and thus becomes a crank. The stroke line $R\ L$ passes through the center of D, which is below the center of P. The forward, or working, stroke occurs while the crank-pin b passes over the arc $d\ c\ g$, and the quick return occurs while it travels over the arc $g\ e\ d$. During each of these

intervals, the link *C D* completes a half revolution, and, consequently, must move more rapidly, while the crank-pin describes the shorter arc.

1455. To proportion the motion, it is only necessary to so locate the stroke line *R L* that it will divide the crank-pin circle *d c g e* into two parts, *d c g* and *g e d*, in the proportions of the forward and return strokes. The point *D*, where this line cuts the center line *S T*, is the position for the center of the slotted crank. The motion is plotted as in Fig. 360. Divide the crank-pin circle *d c g e* into a number

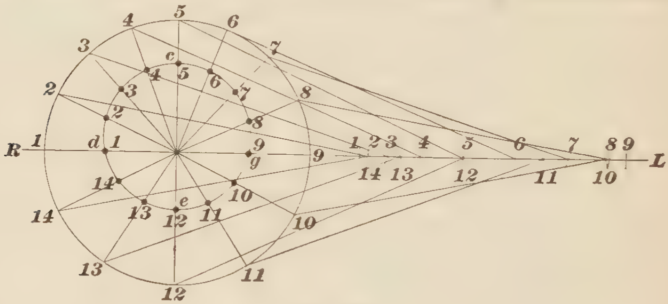


FIG. 360.

of equal parts. From *D*, the center of the slotted crank, draw radial lines through these points to the outer circle, which represents the path of pin *C* (Fig. 359), using the latter points of intersections as centers, and with a radius equal to the length of the connecting-rod, strike off points on the stroke line, which will show the movement of the tool for equal amounts of rotation of the driving gear.

1456. Fig. 361 shows the mechanism as practically constructed. The gear *G* is driven with a uniform velocity, in the direction of the arrow, by the pinion *H*. It rotates upon the large pin *P* (which is a part of the frame of the machine), and carries the pin *b* which turns in the block *k* and is capable of sliding in a radial slot in the piece *B*, as clearly shown in the sectional view. This piece *B* is supported by the shaft *D*, which turns in a bearing extending through the lower part of the large pin. *R L*, drawn through the center of *D*, is the line of motion of the tool slide. The

connecting-rod, actuating the tool slide, is pivoted to the stud C, which is clamped to piece B. The parts are lettered

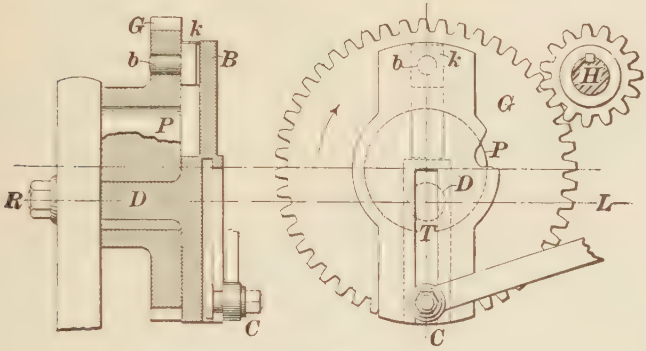


FIG. 361.

as in the two previous cuts, and the student should be able to study out the working of the mechanism without further explanation.

1457. The Adjustment of the Stroke.—The radial slot T, in Fig. 361, to which the connecting-rod is attached, provides for the adjustment of the *length* of the stroke, and if the point of attachment of the rod to the tool slide is also made adjustable, the *position* of the stroke, as well as its length, can be changed. Thus, in the case of a shaping machine, it is not only desirable to regulate the *distance* passed over by the tool, but to have the stroke extend *exactly to a certain point*. These two adjustments are often required in mechanism where reciprocating pieces are employed.

OTHER LINKAGES.

1458. The **universal joint,** shown in Fig. 362, is used to connect two shafts, the center lines of which are in the same plane, but make an angle with one another. It is generally constructed in the following manner: Forks F, F are fastened to the ends of the shafts A and B, and have burrs a, a tapped out to receive the studs S, S. The ends of these studs are turned cylindrical, and are a working fit, in

corresponding bearings in the ring *R*. The details of construction may be seen in the right-hand part of this figure.

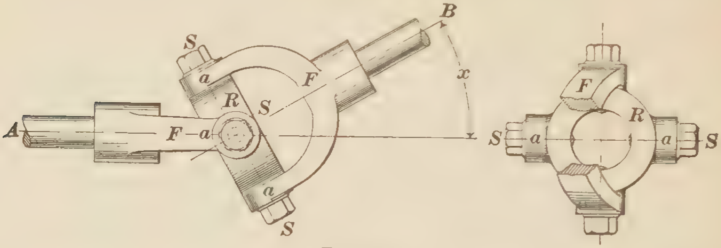


FIG. 362.

In heavy machinery the forks are forged and welded to the shafts.

1459. An objection to the universal joint is that the motion transmitted is not uniform. During one revolution the speed of the driven shaft varies twice between a velocity that is greater, and one that is less than the velocity of the driving shaft, and between these points there are four positions where the velocity of the two shafts is the same. Suppose the driving shaft to revolve uniformly; the *least* speed of the driven shaft will be equal to the speed of the driving shaft, multiplied by the cosine of the angle between the two axes produced (the angle x in Fig. 362). The *greatest* speed will be equal to the speed of the driver, multiplied by $\frac{1}{\cos x}$. Thus, if the shafts revolve at the rate of 100 revolutions per minute, and the angle between them is 30°, the least speed of the driven shaft will momentarily be at the rate of $100 \times \cos 30° = 100 \times .86603 = 86.6$ revolutions per minute, and the greatest speed, $\frac{100}{\cos 30°} = \frac{100}{.86603} = 115.47$ revolutions per minute.

1460. To obviate this trouble, which brings excessive wear and stresses on the working parts, the **double universal joint** is used, as shown in Fig. 363. Let *A* and *B* be the two shafts to be connected. Draw their center lines, intersecting at *o*, and bisect the angle *A o d* by the line *a o*.

The center line $e f$ of the connecting shaft D must now be drawn perpendicular to $a\ o$. Care must be taken that the forks on the intermediate shaft lie in the same plane. Thus constructed, a uniform motion of A will give a varying motion to D, which in turn will transmit to B the same motion as that of A. This arrangement is often employed to connect parallel shafts, as would be the case in the figure if $o\ d$ took the direction $s\ t$. In such a case, it makes no difference what angle is made by $e\ f$, except that, if the joints are expected to wear well, it should not be too great.

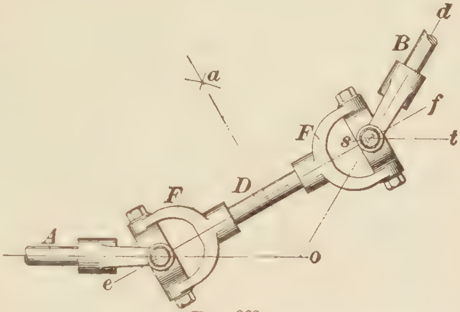


FIG. 363.

1461. Watt's Parallel Motion.—A **parallel motion,** more properly called a **straight-line motion,** is a link mechanism designed to guide a reciprocating piece, as a piston rod, in a straight line. In the early days of the steam engine, parallel motions were extensively used to guide the pump and piston rods, but are now seldom met with, except on steam-engine indicators, where they are employed to give a straight-line motion to the pencil. Very few parallel motions produce an absolutely straight line, and it is customary to design them so that the middle and two extreme positions of the guided point will be in line. The best known motion is the one shown in Fig. 364, which was invented by James Watt, in 1784. The links $A\ B$ and $C\ D$ vibrate on their fixed centers A and D. The other ends, B and C, are connected by the link $C\ B$, which has the point O so chosen that it will pass through three points, O_1, O, and O_2, in the straight line $S\ S$ perpendicular to the links $C\ D$ and $A\ B$ when in their middle positions. When the point O is at the upper extremity of its motion at O_1, the

linkage assumes the position $A\ B_1\ C_1\ D$; at the lower extremity it assumes the position $A\ B_2\ C_2\ D$.

1462. Having given the length $O_1\ O_2$ of the stroke, and O, the middle position of the guided point, the center of one lever as A, and the perpendicular distance between

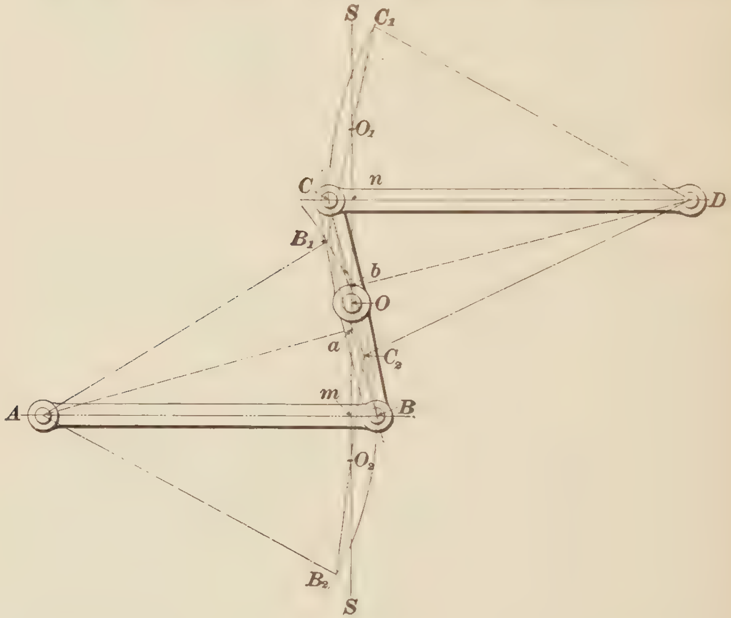


FIG. 364.

the levers when in mid position, the motion may be laid out as follows: Let $S\ S$ be the path of the guided point, O its middle position, A the given center, and $A\ B$ and $C\ D$ indefinite parallel lines, representing the middle positions of the levers. From m, where $A\ B$ intersects $S\ S$, lay off upon $S\ S$ the distance $m\ a$, equal to $\frac{1}{4}$ of the stroke. Join A with a and draw an indefinite line $a\ B$, perpendicular to $A\ a$. The point B, where $a\ B$ intersects line $A\ B$, is the right-hand extremity of lever $A\ B$, and the lower extremity of link $C\ B$. The point C is obtained by drawing an indefinite line through B and O; where it intersects the line $C\ D$ will be the point.

To find center D lay off $n\,b = \frac{1}{4}$ stroke; connect C and b, and from b draw an indefinite line perpendicular to $C\,b$; the center will be at its intersection with $C\,D$.

If the positions of both centers should be known, mark points a and b as before. Draw $A\,a$ and $b\,D$, and through b and a draw perpendiculars to these lines; the points B and C, where they intersect the center lines of the levers, are the extremities of these levers. Join B and C by the link $B\,C$, and the point O, where the center line of this link cuts the line of motion $S\,S$, is the position of the guided point O on the link $B\,C$.

CAMS.

1463. A **cam** is a turning or sliding piece, which, by the shape of its curved edge, or a groove in its surface, imparts a variable or intermittent motion to a roller, lever, rod, or other moving part.

1464. General Case. Fig. 365 represents the general case for a **plate cam.** The cam C is supposed to turn in a right-handed direction about the axis B, and to transmit a variable motion through the roller A, and the lever L, to the rod R. The lever swings on the axis 0_0, the roller moving up or down on the arc 0 to 6, as the cam revolves. The roller is held in contact with the cam by its own weight and that of the lever and the rod.

Suppose the location of the cam shaft B and the rod R to be known, and that the cam is to revolve uniformly right-handed and impart motion to the rod, so that during the first half of every revolution it will move uniformly downwards, during the next quarter turn it will remain stationary, and during the last quarter it will return to its former position with a uniform motion.

1465. In order to determine the outline of the cam upon which the *circumference* of the roller bears, it is necessary to find an outline which will give the *center* of the roller the required motion. Then, by placing the point of the

compasses at different places on this outline, and striking arcs inside of it, with radii equal to the radius of the roller, the curve for the actual cam can be drawn, the curves being tangent to the arcs, and parallel to the first outline.

In this case, the roller *A* moves in an arc directly over the center of *B*. Knowing the distance the rod *R* is to move, we must so choose the point O_0 and the throw of the cam, that is, the distance that *A* is to move, that (see figure) the movement of *R* will be to the movement of *A* as *r* is to *a*.

Now, with O_0 as a center and a radius equal to *a*, describe

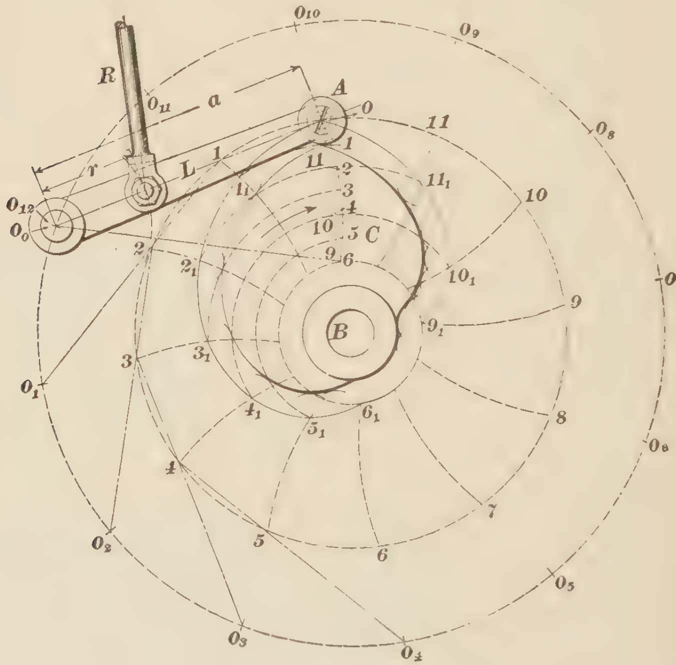


FIG. 365.

the arc *0-3-6*, in which the center of the roller is to move, and mark the highest and lowest points *0* and *6* of the roller. The lower point should not be near enough the shaft to allow the roller to strike the hub of the cam.

1466. It evidently makes no difference with the relative motions of the cam and roller whether the cam turns

right-handed, and the lever remains with its axis at O_0, or whether the cam is assumed to be stationary, and the lever and roller move *left-handed* in a circle about the center B. This latter process will be adopted.

With B as the center, draw a circle through O_0 and space it into a number of equal parts, say 12, and number the divisions around to the left. Now, assume the lever to move around the axis B in a left-handed direction. It will take positions O_1-*1*, O_2-*2*, O_3-*3*, etc. Hence, using these several points on the outer circle as centers, and with radii equal to a, the length of the lever, describe a series of arcs corresponding to the original arc *0-3-6*. Number these arcs *1*, *2*, *3*, etc., to correspond with the numbers on the outer circle.

During the first half-turn of the cam, or, what is the same thing, while the lever is moving from its position at O_0 to O_6 on the outer circle, the center of the roller must move uniformly from its outer to its inner position. Hence, draw the chord of the original arc *0-3-6*, and divide it into six equal parts, numbering them towards the center as shown. Then, with B as a center, describe an arc through point *1*, intersecting arc No. *1* in point 1_1. Now, sweep arcs through the other points, getting 2_1, 3_1, 4_1, etc., which are all points in the curve of the cam outline, for the center of the roller. From O_6 to O_9 (on the original circle), the center of the roller remains at a constant distance from B; hence, *6* and *9* must be connected by a circular arc. From O_9 to O_{12} the points are found as before by dividing the chord *0-6* into three equal parts and numbering them as shown, the numbers running outwards.

The final steps are to draw the cam outline for the center of the roller through points 1_1, 2_1, 3_1, 4_1, etc.; then, draw the outline for the cam itself parallel to it, as explained at first. This is easily done by setting the dividers to describe a circle whose radius shall be the same as that of the roller A. Then, with various points on the curve 0-1_1-2_1-3_1..........11_1 as centers (the more, the better), describe short arcs as shown. By aid of the irregular curve, draw a curve which shall be tangent to the series of short arcs; it will be the required

outline of the cam, and will be parallel to the curve $0\text{-}1_1\text{-}2_1\text{-}3_1 \ldots\ldots\ldots\ldots 11_1$.

1467. The question sometimes arises in designing cams of this nature, whether it is the chord *0-6* or the arc *0-6* that should be divided to give the roller the proper outward and inward motions. For all practical purposes either way is sufficiently exact, but *neither* is quite correct, though it is better to space the chord. The *exact* way would be to draw the rod R and the roller in the different positions desired, and then design the cam to meet the roller at these points.

1468. The cam shown in Fig. 366 differs in principle from the preceding one only in that the roller moves in a straight line, *passing to one side* of the center B of the shaft. Let it be required to design a cam of this nature to revolve right-handed, and which shall cause the roller A and rod R to rise with a uniform motion to a distance h during two-thirds of a revolution. When the roller reaches its highest point, it is to drop at once to its original position, and to remain there during the remainder of the revolution. Assume the distance from the center B to the center line of R to be equal to r.

With r as a radius, describe a circle about B, as shown. The center line of the rod will be tangent to this circle in all positions. With the same center and a radius equal to $B\,A_0$, A_0 being the extreme outward position of the roller, describe the outside circle A_0 *4-8* A_0. Divide this circle into some convenient number of equal parts, the number depending upon the fraction of a revolution required for the different periods of motion. Since the roller is to rise during two-thirds of a revolution, we may use 12 divisions as before, thus giving $\frac{2}{3} \times 12 = 8$ whole divisions for the first period.

Now, proceeding as before, by assuming the rod to move about the cam to the left, its positions when at the points of division A_1 A_2, etc., will be represented by drawing lines

through these points, and tangent to the inner circle, whose radius is r. A and A_0 are the two extreme positions of the roller. Divide the line $A\ A_0$ into eight equal parts, numbering them from the inside outwards, since the first movement of the roller is outwards. With B as a center, draw concentric arcs through these points, intersecting the tangents at 1_1, 2_1, 3_1, etc. At point 8 on the outer circle, the roller drops along the line 8-8_1, the point 8_1 being determined by drawing an arc about B with a radius $B\ A$. From

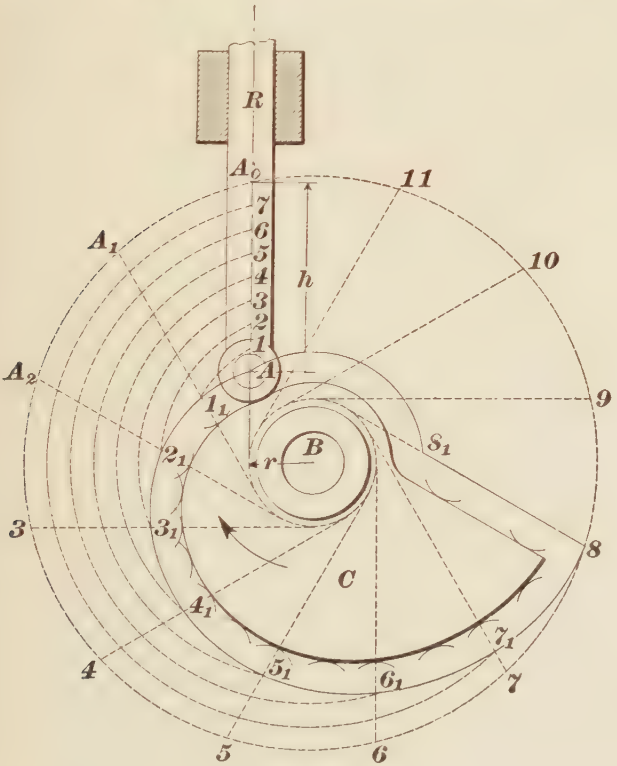


FIG. 366.

point 8 back to A the rod is at rest. The true cam outline is now to be found, as was done in the last example, by striking small arcs from points on the curve $A\ 1_1$-2_1-3_1-4_1-5_1-6_1-7_1-8-8_1

as centers. This cam can revolve in only one direction when operating the rod *R*.

1469. Harmonic-Motion Cams.—If a cam is required to give a rapid motion between two points, without regard to the *kind* of motion, its surface should be laid out so as to gradually accelerate the roller at the start, and to gradually retard it at the end of its motion, in order that the movement may be as smooth and free from shocks as possible. For this reason cams are frequently designed to produce harmonic motion—that is, a uniform motion of the cam produces a motion of the roller like that of the slotted cross-head in Fig. 350. The diagram in Fig. 367 shows how this latter motion is plotted. Let *A C* be the stroke line of

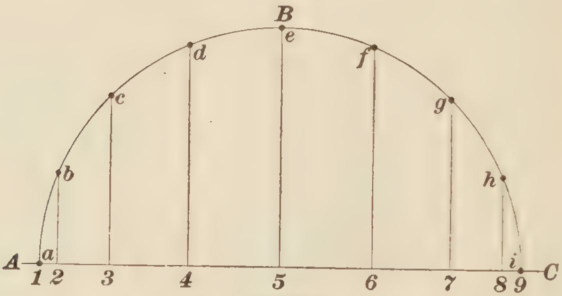


FIG. 367.

the cross-head, and *A B C* the crank-pin circle, which is divided into a number of equal spaces by the points *a*, *b*, *c*, *d*, etc. Dropping perpendiculars from these points, we obtain points *1*, *2*, *3*, *4*, etc., on the stroke line. The spaces between the latter points represent the distances traversed by the cross-head while the crank-pin moves through the equal spaces *a b*, *b c*, *c d*, etc. It will be seen that the distances increase from points *1* to *5*, and decrease from *5* to *9*.

1470. To apply the motion to the two cams previously taken up, we should simply have to lay off the distances *1-2*, *2-3*, *3-4*, etc., on the chord *0-6*, in Fig. 365, or the line $A_0 A$, in Fig. 366, in place of the equal spaces used in these figures.

Fig. 368, which represents the left side of the first cam considered, laid out in this way, shows a convenient method of spacing. Upon the chord *0-6*, as a diameter, draw the semi-circle *0 d 6*; divide it into a suitable number of equal parts and project these divisions by straight lines to the chord *0-6*. Through the points of intersection *1*, *2*, *3*, *4*, etc., and with *B* as a center, describe the arcs $1\text{-}1_1$, $2\text{-}2_1$, $3\text{-}3_1$, etc., and complete the cam outline, as before.

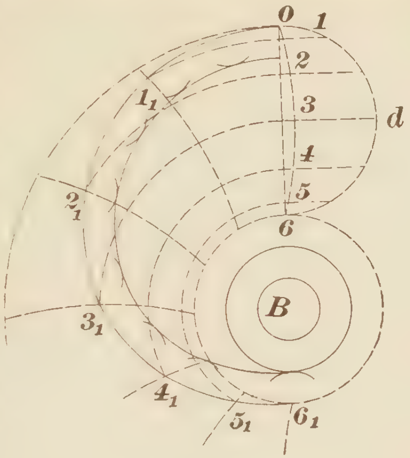


FIG. 368.

1471. Positive-Motion Cams.—The cams thus far considered can drive the roller in one direction only, making a spring or weight necessary to keep the two in contact. If the cam plate should extend beyond the roller, however, and a groove should be cut in it for the roller to run in, the motion of the roller would be *positive* in both directions.

1472. The word **positive,** when applied to a mechanism, has a different meaning from any heretofore given to it. A mechanism so constructed that nothing short of actual breakage of some one of its parts can keep it from working properly when motion is imparted to one of the links which operates it is called a **positive mechanism,** or a **positive gear,** when speaking of valve gears, and the motion produced is called a **positive motion.** Those mechanisms which depend for their operation upon the raising or lowering of a weight (i. e., upon gravity), or the action of a spring, are termed **non-positive** or **force-closed mechanisms.** Non-positive mechanisms, although extensively used, are of a lower order of mechanical excellence than positive mechanisms, and, other things being

equal, a positive motion should be chosen when designing a mechanism, since a non-positive one will refuse to work if the weight or the part operated by the spring should get "caught."

1473. Sometimes a positive motion is secured with a plate cam by causing it to revolve between two rollers rigidly connected, as illustrated in Fig. 369. The rollers *A* and *E* turn upon pins in one end of the rod *R*. The rod is slotted between the rollers, so that the cam shaft may

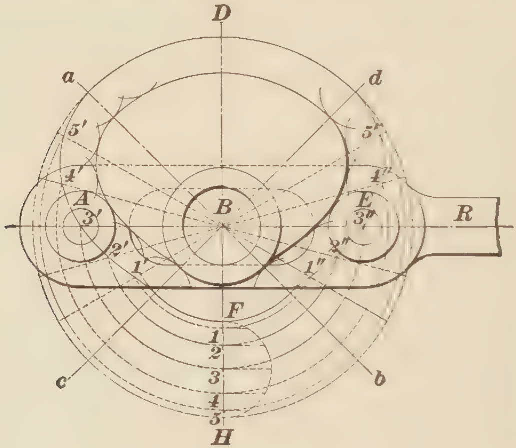


FIG. 369

pass through it and still allow the rod to move to the right and left. The center line of motion of the rollers passes through the center *B* of the shaft, whatever the position of the cam or rod.

1474. Let it be required that during one-quarter of a revolution of the cam the rod shall move to the right or left, according to the position of the cam at the start; that during the next quarter the rod shall be held stationary; that during the third quarter it shall move to its original position, where it is to remain for the rest of the revolution.

Draw the line *D H*, and mark the point *B*. Lay off a distance *B F*, such that when the center of one of the rollers is

at F the hub of the cam will not interfere with the roller. Lay off the distance $F H$ beyond F, equal to the required movement of the rod. With B as a center and a radius equal to $B H$, describe a circle. Divide the circle into four quadrants by the lines $a b$ and $c d$, making angles of 45° with $D H$, and with center B and radius $B F$ strike an arc as shown. That part of the arc included in the lower quadrant, and that part of the outer circle included in the upper quadrant, form two parts of the required outline, and the distance between the centers of the rollers must be equal to $D F$.

Now, for the two side quadrants, it is evident that the distance between any two diametrically opposite points on the outline must be equal. Thus, the distance from $1'$ to $5''$, from $2'$ to $4''$, from $3'$ to $3''$, must all be equal to the distance $D F$. These points are obtained by laying off points on $F H$ and drawing arcs through them, proceeding exactly as with the other cams; *but, in order to have the curves correct, point 1 must be as far from F as 5 is from H; point 2 as far from F as 4 is from H, etc.* The harmonic motion curve fulfils this condition, and was used in this case. The other points of construction should be understood from what has gone before.

1475. A third kind of positive-motion cam consists of a cylinder having a groove on the surface, which imparts motion to the roller in a plane parallel to the axis of the cam.

Suppose that during one-half a revolution of the cylinder, in Fig. 370, the arm is to vibrate to the left and back once, as indicated, and that during the other half revolution it is not to move. Let the motion of the roller be harmonic.

The problem consists in finding the center line of the groove, from which, by striking arcs, the sides against which the roller bears can be determined. To lay out the curves, assume the surface of the cylinder to be unwrapped, or developed, as represented by the figure $a b d c$, which represents only a little more than one-half of the length of the

surface, in order to save room. Draw a line $M N$ through the center of this strip, of a length equal to the circumference of the cylinder, and divide it into a number of equal parts, as indicated by O_0 O_1, O_1 O_2, etc. Also, draw two lines, $S T$ and $L P$, parallel to $M N$, and at a distance from it equal to one-half the desired stroke of the roller. Now, with

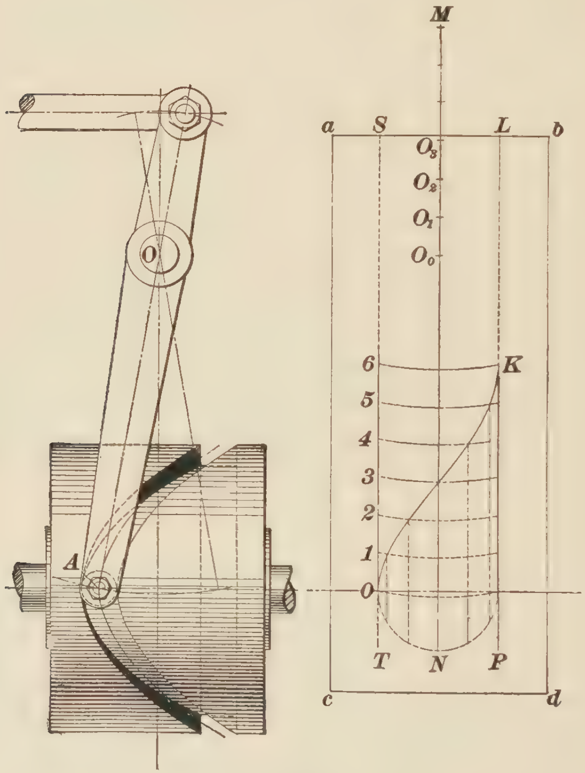


FIG. 370.

a radius $O A$ (left-hand figure), and centers O_0, O_1, O_2, etc., strike arcs *0*, *1*, *2*, *3*, etc., and plot the curve by the aid of these arcs. The line $O K L$ is the center line curve for one-half of the cam, the other half being like the first. The outline can easily be transferred to the cylinder itself.

BELTING.

1476. Belts running over pulleys form a convenient means for transmitting power, but they are not suited to transmit a precise velocity ratio, owing to their tendency to stretch and slip on the pulleys. For driving machinery, however, this freedom to stretch and slip is an advantage, since it prevents shocks that are liable to occur when a machine is thrown suddenly into gear, or when there is a sudden fluctuation in the load.

1477. Velocity Ratio.—Let four pulleys be connected by belt, as shown in Fig. 371. Let D_1 and D_2 be the diameters of the drivers, F_1 and F_2 of the followers, N_1 and N_2

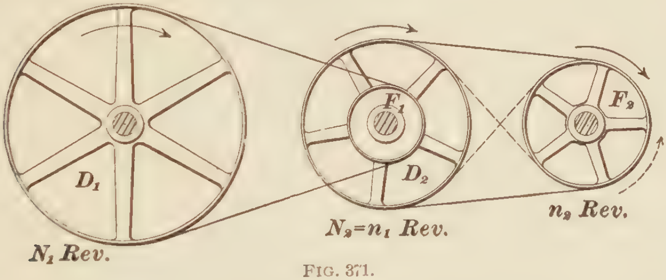


FIG. 371.

the number of revolutions per minute of the drivers, and n_1 and n_2 of the followers. The two middle pulleys are keyed to the same shaft and revolve together.

Consider first the pulleys whose diameters are D_1 and F_1. Assuming that there is no slipping or stretching of the belt, the circumferential speeds of the pulleys will be the same as the velocity of the belts passing over them. Hence, $D_1 \times 3.1416 \times N_1$, the circumferential speed of the first driver, $= F_1 \times 3.1416 \times n_1$, the circumferential speed of the first follower. Canceling 3.1416 from both sides of the equation, we have $D_1 N_1 = F_1 n_1$, or, dividing by D_1 and n_1,

$$\frac{N_1}{n_1} = \frac{F_1}{D_1}. \qquad \textbf{(136.)}$$

That is, *the speeds or numbers of revolutions of two connected pulleys are inversely proportional to their diameters.*

1478. A short way of applying this principle is by the following rule for two pulleys:

Rule.—*Multiply together the number of revolutions and diameter of one pulley, and divide by the given number of revolutions, or given diameter, of the other pulley. The result will be the required diameter or number of revolutions.*

EXAMPLE.—A pulley 30 inches in diameter, making 210 revolutions per minute, drives a second pulley 14 inches in diameter. How many revolutions per minute does the latter pulley make?

SOLUTION.—$30 \times 210 = 6,300$, and $6,300 \div 14 = 450$ revolutions. Ans.

EXAMPLE.—The driving pulley of a machine is one foot in diameter and must make 750 revolutions in 5 minutes. What size pulley should be used on the driving shaft, if its speed is 143 revolutions per minute?

SOLUTION.—In all examples of this kind the speeds and diameters must be reduced to the same units. 750 rev. in 5 min. $= 750 \div 5 = 150$ rev. per min.; one foot $= 12$ in. Hence, $12 \times 150 = 1,800$, and $1,800 \div 143 = 12.6$ in., nearly. Ans.

1479. From the equation $D_1 N_1 = F_1 n_1$, derived above, we obtain, by dividing by F_1, $n_1 = \frac{D_1 N_1}{F_1}$. In like manner, taking the other two pulleys in Fig. 371, we obtain $N_2 = \frac{F_2 n_2}{D_2}$. But the two middle pulleys revolve together, so that the values of N_2 and n_1 are equal, and may be placed equal to each other; thus, $\frac{D_1 N_1}{F_1} = \frac{F_2 n_2}{D_2}$, or, multiplying by F_1 and D_2, $N_1 D_1 D_2 = n_2 F_1 F_2$. **(137.)**

1480. That is, *the speed of the first pulley, multiplied by the diameter of each of the drivers, equals the speed of the last pulley, multiplied by the diameter of each follower.*

This formula is in most cases convenient to apply as it stands.

EXAMPLE.—Referring to Fig. 371, let the diameters of the drivers be 32 in., the diameter of the first follower be one foot, and of the second follower 15 inches. If the first shaft has a speed of 60 revolutions per minute, what is the speed of the last shaft?

SOLUTION.—Substituting in formula **137**, $60 \times 32 \times 32 = n_2 \times 12 \times 15$, or dividing by 12×15, $n_2 = \frac{60 \times 32 \times 32}{12 \times 15} = 341\frac{1}{3}$ rev. per min. Ans.

EXAMPLE.—An emery grinder is to be set up to run at 1,200 revolutions per minute. The countershaft (corresponding to the middle shaft in Fig. 371) has pulleys 20 and 8 inches in diameter. If the pulley on the grinder is 6 inches in diameter, what size pulley must be used on the main line shaft, its speed being 180 revolutions per minute?

SOLUTION.—Substituting in formula **137**, $180 \times D_1 \times 20 = 1{,}200 \times 8 \times 6$, or $D_1 = \frac{1{,}200 \times 8 \times 6}{180 \times 20} = 16$ inches. Ans.

1481. When the speeds of the first and last shafts are given, and the diameters of all the pulleys are to be found, the following method is convenient:

Rule.—*Divide the higher speed by the lower. If two pulleys are to be used, this will be the ratio of their diameters.*

If four pulleys are required, find two numbers whose product equals the above quotient. One of these numbers will be the ratio of the diameters of one pair of pulleys, and the other number of the other pair.

EXAMPLE.—It is required to run a machine 1,600 revolutions per minute, the driving shaft making 320 revolutions per minute. What size pulleys are required, (*a*) when two pulleys are used; (*b*) when four pulleys are used?

SOLUTION.—(*a*) $1{,}600 \div 320 = 5$. The two pulleys must, therefore, be in the ratio of 5 to 1, the driving pulley being 5 times as large as the driven pulley, since the latter has the greater speed. We will assume diameters of 30 and 6 inches.

(*b*) $2\frac{1}{2} \times 2 = 5$. One pair of pulleys must be in the ratio of $2\frac{1}{2}$ to 1, and the other pair of 2 to 1. We will assume diameters of 25 and 10 inches for one pair, and of 12 and 6 inches for the other pair.

1482. Direction of Rotation.—It will be noticed, by referring to Fig. 371, that the pulleys are connected by **open belts** where indicated by full lines, and by **crossed belts** where indicated by dotted lines. *Pulleys connected by open belts turn in the same direction, and those connected by crossed belts in opposite directions.*

EXAMPLES FOR PRACTICE.

1. A driving pulley is 54 inches in diameter, and a driven pulley which runs at 112 revolutions per minute is $2\frac{1}{2}$ feet in diameter. What is the speed of the driving shaft? Ans. 62.22 rev. per min.

2. The fly-wheel of an engine running at 180 revolutions per minute is 8 ft. 5 in. in diameter. What should be the diameter of the pulley

which it drives, if the required speed of the latter is 600 revolutions per minute? Ans. $30\frac{5}{16}$ in., nearly.

3. In Fig. 371, let the diameters of the two drivers be 48 and 25 inches, and of the two followers 16 and 12 inches. If the last driver's shaft rotates 800 times in 5 minutes, what is the speed per minute of the first shaft? Ans. 25.6 rev. per min.

4. If the first pulley in Fig. 371 turns right-handed, and is connected with the second by a crossed belt, and the third with the fourth by an open belt, in what direction would the last pulley turn? In what direction would it turn if two crossed belts were used?

5. A machine is to be belted through a countershaft, so as to run at 1,200 revolutions per minute, the speed of the driving shaft being 120 revolutions per minute. Find three ratios that could be used for each pair of pulleys.

Ans. $\begin{cases} 5 : 1 \text{ and } 2 : 1. \\ 4 : 1 \text{ and } 2\frac{1}{2} : 1. \\ 3\frac{1}{3} : 1 \text{ and } 3 : 1. \end{cases}$

6. An emery grinder is to be set to run at 1,400 revolutions per minute. The countershaft has pulleys 30 and 8 inches in diameter. The pulley on the grinder is 7 inches in diameter. What size pulley should be used on the main line shaft, its speed being 185 revolutions per minute? Ans. $14\frac{1}{8}$ inches.

POWER TRANSMISSION BY BELT.

1483. The Effective Pull.—In Fig. 372, let D and F be two pulleys connected by a belt, D being the driver and F the follower. To avoid undue slipping, the belt must be drawn tight. This will produce a tension in the upper and lower parts which we call T_2 and T_1, respectively.

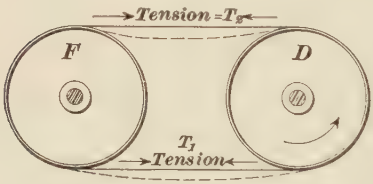


Fig. 372.

Suppose the two pulleys to be stationary and that the belt is put on with a certain tension. Then, T_1 will equal T_2. But if the pulley D should be turned in the direction of the curved arrow it would tend to stretch the lower part of the belt, increasing its tension still more, while the tension of the upper part would be diminished an equal amount. This would go on until the difference of the tensions was sufficient to start pulley F.

This difference ($T_1 - T_2$) is the pull that does the work in transmitting power, and is called the effective pull. In any

case, therefore, the number of foot-pounds of work transmitted by a belt must equal the effective pull of the belt in pounds times the number of feet passed through; or, taking a minute as the unit of time, *the power transmitted in foot-pounds per minute = the effective pull × the velocity in feet per minute.*

1484. To show clearly how the effective pull enters into calculations of power transmissions, two examples will be solved.

EXAMPLE.—The diameter of the driving pulley D is 36 inches. It makes 150 revolutions per minute and carries a belt transmitting 6 horsepower. What is the effective pull of the belt?

SOLUTION.—Velocity of the belt in feet per minute =

$$\frac{150 \times 36 \times 3.1416}{12} = 1{,}413.72.$$

This, multiplied by the effective pull in pounds must equal the foot-pounds of work done per minute, or $6 \times 33{,}000$. Hence, letting P = the effective pull, and equating these expressions, we have

$$P \times 1{,}413.72 = 6 \times 33{,}000 = 198{,}000,$$

or $P = \dfrac{198{,}000}{1{,}413.72} = 140.06$ lb., nearly.

EXAMPLE.—A pulley 4 feet in diameter is driven at 100 revolutions per minute, and transmits power to another pulley by means of a belt without slip. If the tension on the driving side of the belt is 400 pounds, and on the slack side is 100 pounds, what is the horsepower transmitted?

SOLUTION.—$P = (T_1 - T_2) = 400 - 100 = 300$ lb. Horsepower transmitted $\times$ 33,000 = foot-pounds of work done per minute; or

$$\text{H.P.} \times 33{,}000 = 300 \times 100 \times 4 \times 3.1416, \text{ whence}$$

$$\text{H. P.} = \frac{300 \times 100 \times 4 \times 3.1416}{33{,}000} = 11.424. \quad \text{Ans.}$$

1485. To Determine the Width of Belt.—A belt should be wide enough so that it will bear safely and for a reasonable length of time the greatest tension that will be put upon it. This will be the tension T_1 of the driving side of the belt. As belts are usually laced, or fastened with metallic fasteners, both of which require holes to be punched in the ends, it is customary to use the breaking strength through the lace holes, divided by a suitable factor of safety, as the greatest allowable tension. The average breaking strength for single leather belts, through the lace

holes, is 200 pounds per inch of width. This divided by three, which is a suitable factor of safety for belting, gives $66\frac{2}{3}$ pounds. Thus, in the last example, the tension of the driving side of the belt was assumed to be 400 pounds. Hence, using $66\frac{2}{3}$ pounds as the safe working stress per inch of width, a belt $\frac{400}{66\frac{2}{3}} = 6$ inches wide would be required.

This tension T_1, for any particular case, depends upon three things—viz., the effective pull of the belt, the coefficient of friction between the belt and pulley, and the size of the arc of contact of the belt on the smaller pulley. As the equations involving these quantities are somewhat complicated, Table 33 has been calculated. It will afford a convenient means for finding not only the width of belt for a given horsepower, but the horsepower for a given width as well. In the first column, the arc covered by the belt is stated in degrees, and in the second column in fractional and decimal parts of the circumference covered. The third column gives the greatest allowable values of $(T_1 - T_2)$, or the effective pull, per inch of width, for single leather belts having any arc of contact. It was computed by assuming a value for T_1 of $66\frac{2}{3}$ pounds, and a coefficient of friction of .27. This latter has been found by experiment to be a fair value to use for leather belts running over cast iron pulleys, under conditions met with in practice.

TABLE 33.

Arc Covered by Belt. Degrees.	Arc Covered by Belt. Fraction of Circumference.	Allowable Value of Effective Pull, or $(T_1 - T_2)$ per Inch of Width.
90	$\frac{1}{4} = .25$	23.0
$112\frac{1}{2}$	$\frac{5}{16} = .312$	27.4
120	$\frac{1}{3} = .333$	28.8
135	$\frac{3}{8} = .375$	31.3
150	$\frac{5}{12} = .417$	33.8
$157\frac{1}{2}$	$\frac{7}{16} = .437$	34.9
180 or over.	$\frac{1}{2} = .50$	38.1

1486. To use the table in finding the width of a single leather belt required for transmitting a given horsepower, we have the following rule :

Rule.—*Compute the effective pull of the belt. Divide the result by the suitable effective pull per inch of width, as given in Table 33; the quotient will be the width of belt required, in inches.*

EXAMPLE.—What width of single belt is needed to transmit 20 horsepower with contact on the small pulley of $\frac{3}{8}$ of the circumference and a speed of 1,500 feet per minute?

SOLUTION.—First finding the effective pull, $P \times 1{,}500 = 20 \times 33{,}000$,

$$P = \frac{20 \times 33{,}000}{1{,}500} = 440.$$

Hence, $440 \div (T_1 - T_2)$, from the table, $= \frac{440}{31.3} = 14$ inches, nearly. A 14-inch belt would be used. Ans.

1487. To Determine the Horsepower that a Belt Will Transmit.—The process for a single belt must evidently be just the reverse of the preceding. It is as follows:

Rule.—*Muliply together a suitable value for the effective pull, taken from Table 33, the width of the belt in inches, and its velocity in feet per minute. The result divided by 33,000 will be the horsepower that the belt will transmit.*

EXAMPLE.—What horsepower will a one-inch belt transmit with a speed of 900 feet per minute and an arc of contact of 180°?

SOLUTION.—$T_1 - T_2$, from the table, $= 38.1$. $38.1 \times 1 \times 900 = 34{,}290$, which, divided by 33,000, gives 1.04 horsepower, nearly. Ans.

1488. General Rule for Belting.—From the last example, we see that *a single belt traveling 900 feet a minute will transmit one horsepower per inch of width when the arc of contact on the smaller pulley does not vary much from 180°.* This is used by many engineers as a general rule for belting, to be applied to all cases.

The following three formulas express the operations that would be performed in applying this rule:

Let H = horsepower to be transmitted;
W = width of belt in inches;
S = belt speed in feet per minute.

Then,
$$H = \frac{WS}{900}. \qquad \textbf{(138.)}$$
$$W = \frac{900\,H}{S}. \qquad \textbf{(139.)}$$
$$S = \frac{900\,H}{W}. \qquad \textbf{(140.)}$$

EXAMPLE.—Two pulleys, 48 inches in diameter, are to be connected by a single belt, and make 200 revolutions per minute. If 40 horsepower is to be transmitted, what must be the width of belt?

SOLUTION.—The belt speed $= \frac{200 \times 48 \times 3.1416}{12} = 2{,}513$ feet per minute, about. Applying formula **139**, $W = \frac{900 \times 40}{2{,}513} = 14.3$ inches. Ans.

A 14-inch belt might safely be used, since the rule gives a liberal width when the pulleys are of equal size.

EXAMPLE.—What size pulleys should be used for a 4-inch belt, which is to connect two shafts running at 400 revolutions per minute and transmit 14 horsepower? Both pulleys are of the same size.

SOLUTION.—By formula **140**, $S = \frac{900 \times 14}{4} = 3{,}150$ feet per minute. Since this speed = the circumference of the required pulley in feet $\times$ 400, we have circumference of pulley $= \frac{3{,}150 \times 12}{400} = 94.5$ inches; diameter $= \frac{94.5}{3.1416} = 30$ inches. Ans.

1489. Double Belts.—Double belts are made of two single belts cemented and riveted together their whole length, and are used where much power is to be transmitted. As the formulas for single belts are based upon the strength through the rivet holes, a double belt, which is twice as thick, should be able to transmit twice as much power as a single belt, and, in fact, more than this, where, as is quite common, the ends of the belt are glued instead of being laced.

Where double belts are used upon small pulleys, however, the contact with the pulley face is less perfect than it would be if a single belt were used, owing to the greater rigidity of the former. More work is also required to bend the belt as it runs over the pulley than in the case of the thinner and more pliable belt, and the centrifugal force tending to throw the belt from the pulley also increases with the thickness. Moreover, in practice, it is seldom that a double belt is put on with twice the tension of a single belt. For these reasons, the width of a double belt required to transmit a given horsepower is generally assumed to be seven-tenths the width of a single belt to transmit the same power. Upon this basis, formulas **138, 139,** and **140** become, for double belts,

$$H = \frac{WS}{630}. \qquad \textbf{(141.)}$$

$$W = \frac{630\,H}{S}. \qquad \textbf{(142.)}$$

$$S = \frac{630\,H}{W}. \qquad \textbf{(143.)}$$

EXAMPLES FOR PRACTICE.

1. If the effective pull on a belt per inch of width is 50 pounds, and the belt passes over a pulley 36 inches in diameter, which makes 160 revolutions per minute, how wide should the belt be to transmit 12 horsepower? Ans. $5\frac{1}{4}$ inches.

2. What width of single belting should be used to transmit 5 horsepower, when the belt speed is 2,000 feet per minute, and the arc of contact on the smaller pulley is 90°? Ans. $3\frac{1}{2}$ + inches.

3. Using the general rule, find the horsepower that a 16-inch single belt will transmit, the belt speed being 1,000 feet per minute. Ans. 17.8 H. P.

4. Calculating from Table 33, how much power could the above belt be depended upon to transmit if the arc of contact on the smaller pulley is $\frac{1}{3}$ of the circumference? Ans. 14 H. P., nearly.

5. Required the diameter of pulleys necessary to enable a 10-inch belt to transmit 9 horsepower at 125 revolutions per minute, both pulleys being of the same size. Ans. 2 ft., nearly.

6. How much power would the belt in example 3 transmit, if the belt were double? Ans. 25.4 H. P., nearly.

SPEED CONES.

1490. Speed cones are used for varying the velocity of a shaft or other rotating piece driven by a belt. Their method of operation will be clearly seen in Figs. 373 to 376.

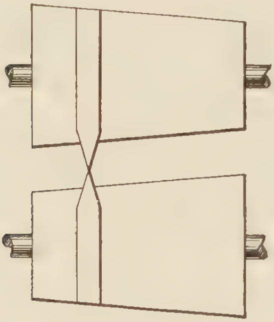


FIG. 373.

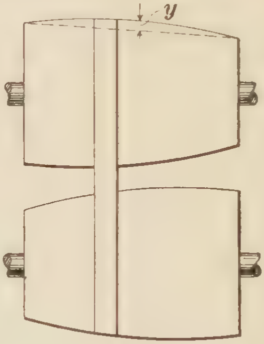


FIG. 374.

Figs. 373 and 374 show continuous cones and conoids, respectively, the former being suitable for crossed, and the latter for open belts, where the speed of the driver shaft can be raised gradually by shifting the belt. Figs. 375 and 376 show sets of stepped pulleys. As flat belts tend to climb

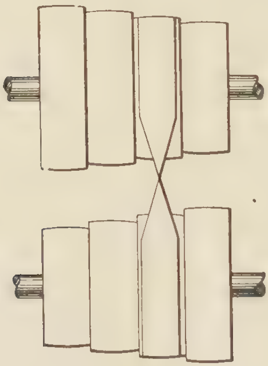


FIG. 375.

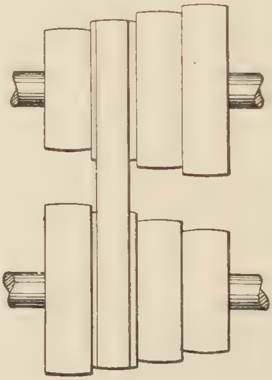


FIG. 376.

a conical pulley, continuous cones or conoids require special provision for keeping the belt in place. For this reason the stepped cones are generally used. Whenever

possible, it is desirable to have both pulleys alike, so that they can be cast from one pattern, and in what follows it will be assumed that this is to be done.

1491. Continuous Cones for Crossed Belts.—Let A and B, in Fig. 377, represent two speed cones of the same size, having the diameters of the large and small ends equal to D and d, respectively. A is the driver, and revolves at a constant number of revolutions N, while the speed of B is n_1 or n_2, according as the belt runs at the small or large end.

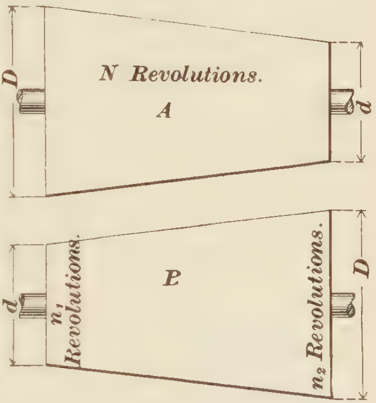


FIG. 377.

It is assumed that pulley B is to have a range of speeds between n_1 and n_2 revolutions, n_1 being the greater.

Since A and B are to be of the same size, it will be found that a certain relation must exist between N and n_1 and n_2 as follows:

From formula **136**,

$$ND = n_1 d, \text{ or } D = \frac{n_1 d}{N}, \qquad \textbf{(144.)}$$

$$\text{and } Nd = n_2 D, \text{ or } D = \frac{Nd}{n_2}. \qquad \textbf{(145.)}$$

Equating **144** and **145**, $\frac{n_1 d}{N} = \frac{Nd}{n_2}$, or

$$N = \sqrt{n_1 n_2}. \qquad \textbf{(146.)}$$

That is, N must equal the square root of the product of n_1 and n_2.

1492. Having determined N, the relation between E and d can be found. From **144** and **145** we have:

$$N = \frac{n_1 d}{D} \text{ and } N = \frac{n_2 D}{d}.$$

Equating, $\frac{n_1 d}{D} = \frac{n_2 D}{d}$, or $n_1 d^2 = n_2 D^2$.

Hence, $$D = d\sqrt{\frac{n_1}{n_2}}. \qquad \textbf{(147.)}$$

That is, the large diameter must equal the small diameter, multiplied by the square root of the quotient of n_1 divided by n_2.

1493. Speed cones, to be properly designed, should have their diameters at different points, so proportioned that the belt will always have the same length, when tightly drawn, whatever its position. Let d and D' represent the diameters of two pulleys, connected by a crossed belt, whose centers are O and C, respectively. (See Fig. 378.) The length of the belt is $2\,A\,B$ + the length of the arcs from F to A and B to E. These arcs subtend equal angles,

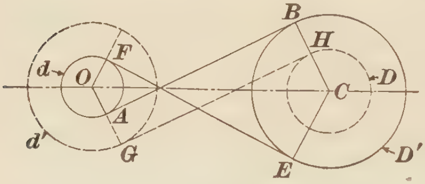


FIG. 378.

and supposing each to contain x degrees, the length of the belt $= 2\,A\,B + 3.1416\,(d + D')\,\frac{x}{360}$, there being 360 degrees in a circle. Now, draw a line $G\,H$ parallel to $A\,B$, and, about O and C as centers, describe dotted circles whose diameters are d' and D tangent to $G\,H$, representing two other pulleys. Then, for the length of the belt, we have $2\,G\,H\ (= 2\,A\,B) + 3.1416\,(d' + D)\,\frac{x}{360}$. But $(d' + D) = (d + D')$, since what was taken from D' to make D was added to d to make d'. Hence, the length of the belt in each case is the same, and we have the rule that *for crossed belts the sum of the corresponding diameters of two speed cones should be the same at all points.*

From this it follows that if two cones not exactly alike are to be driven by cross belt, it is only necessary to see that the sum of the diameters remains the same.

EXAMPLE.—Two continuous speed cones are to be designed to give a range of speed between 100 and 700 revolutions per minute. They are both to be alike in all respects. What must be the speed of the driving shaft, and the large diameter of the cones, assuming the small diameter to be 4 inches?

SOLUTION.—From formula **146**, $N = \sqrt{n_1 n_2} = \sqrt{100 \times 700} = \sqrt{70{,}000} = 264.57 -$ revolutions per minute. Ans.

From formula **147**, $D = d\sqrt{\frac{n_1}{n_2}} = 4\sqrt{\frac{700}{100}} = 4 \times 2.645 = 10.58$ inches. Ans.

1494. When an open belt is used, the values of N, D, and d are calculated as above, but for the other diameters a different rule is required. In Fig. 379, which is similar to Fig. 378, the pulleys are connected by an open belt. The figure is drawn so that $D + d = D' + d'$, the circles D' and d' being made equal to represent the middle sections of two cones. It is evident that the belt over D and d is longer than the one over D' and d'. Hence, we see that the middle sections of speed cones for open belts must be larger proportionately than for crossed belts. This is indicated by y in Fig. 374. Calling this middle diameter M, and letting C be the distance in inches between the centers of the two shafts, upon which the cones are placed, it can be shown that

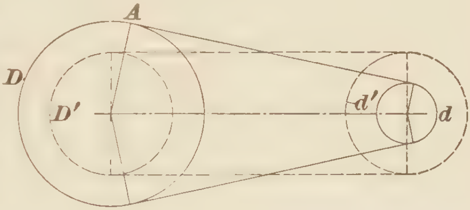


FIG. 379.

$$M = \frac{D + d}{2} + \frac{.08\,(D - d)^2}{C}. \qquad \textbf{(148.)}$$

As the proof is a long one it will not be given. Having thus determined the end and middle diameters, the curve of the conoid may be taken as the arc of a circle passing through the extremities of the three diameters.

EXAMPLE.—What should be the middle diameter for the speed cone of the last example, having end diameters of 4 and 10.58 inches, when an open belt is to be used? The distance between centers is 50 inches.

SOLUTION.—From formula **148**,

$$M = \frac{10.58 + 4}{2} + \frac{.08\,(10.58 - 4)^2}{50}$$
$$= 7.29 + .07 = 7.36 \text{ inches. Ans.}$$

1495. Stepped Pulleys.—When stepped pulleys, or cone pulleys, as they are more commonly called, are to be used, continuous cones should first be laid out as described, like $A\,B\,C\,D$, in Fig. 380. Then draw as many diameters, at equal distances apart, as there are to be steps in the cone plus 1, as $A\ D$, $E\,F$, etc. These will serve as center lines for the different steps, which are to be drawn through the intersections of the above diameters, with the outside lines $A\ B$ and $D\ C$, in the manner indicated.

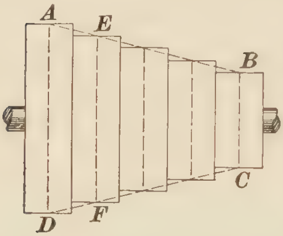


FIG. 380.

1496. It should be noted that, when speaking of a cone pulley as having a certain number of steps, the number of steps is one less than the number of pulleys on the cone. Thus, the cone, in Fig. 380, is a 4-step cone, and has five pulleys. Consequently, if a cone pulley (or cone) is spoken of as having five steps, there are six pulleys and six changes of speed.

If the distance between the axes of the pulleys to be connected by open belt be great, or if, as is sometimes the case, one of the axes be adjustable (the proper tension on the belt being obtained by the weight of the pulley on the adjustable axis), the diameters can be calculated as though the belt were crossed. Otherwise, when designed for open belts they should be laid out as before described.

EXAMPLES FOR PRACTICE.

1. Two continuous speed cones are required to give a range of speed between 100 and 600 revolutions per minute. Assuming the large diameters of the cones to be 14 inches, (*a*) what must be the small diameters, and (*b*) the speed of the driving shaft? Both cones are to be alike.

Ans. { (*a*) 5.71 inches.
(*b*) 244.95 rev.

2. In the above example, if the speed of the driving shaft were 260 revolutions per minute, and the slowest speed of the driven cone 140 revolutions, (*a*) what would be the greatest speed of the driven cone? (*b*) What would be the ratio of the large and small diameters of the cones?

Ans. { (*a*) 482.86 rev.
(*b*) 1.86 : 1.

3. What should be the middle diameter for the speed cones of example 1, when an open belt is used, the distance between centers being 30 inches? Ans. 10.04 inches.

THE CARE AND USE OF BELTING.

1497. Belts most commonly used are of leather, single and double. Canvas belts covered with rubber are sometimes used, especially in damp places, where the moisture would ruin the leather.

Leather belts are generally run with the hair, or grain, side next the pulley. This side is harder and more liable to crack than the flesh side. By running it on the inside the tendency is to cramp or compress it as it passes over the pulley, while, if it ran on the outside, the tendency would be for it to stretch and crack. Moreover, as the flesh side is the stronger side, the life of the belt will be longer if the wear comes upon the weaker or grain side.

1498. The lower side of a belt should be the driving side, the slack side running from the top of the driving pulley. The sag of the belt will then cause it to encompass a greater length of the circumference of both pulleys, as illustrated by the dotted lines in Fig. 372. Long belts, running in any direction other than the vertical, work better than short ones, as their weight holds them more firmly to their work.

It is bad practice to use rosin to prevent slipping. It gums the belt, causes it to crack, and prevents slipping for only a short time. If a belt properly cared for persists in slipping, a wider belt or larger pulleys should be used, the latter to increase the belt speed. Belts, to be kept soft and pliable, should be oiled with castor or neatsfoot oil. Mineral oils are not good for the purpose.

1499. Tightening or guide pulleys, whenever used to increase the length of contact between the belt and pulley or to tighten a belt, should be placed on the slack side, if possible. Thus placed, the extra friction of the guide pulley bearings and the wear and tear of the belt that would result from the greater tension of the driving side are avoided.

1500. Guiding Belts.—When belts are to be shifted from one pulley to another, or must be guided to prevent running off the pulley, the fork, or other device used for guiding, should be close to the driven pulley, and so placed as to guide the advancing side of the belt.

This principle is sometimes made use of where pulleys have flanges to keep the belt on the pulleys. Where constructed with straight flanges, as in Fig. 381, if the belt has any inclination to run on one side, its tendency is to crowd up against the flange as shown at *a*, *a*. When constructed as in Fig. 823,

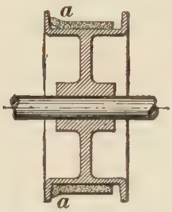


FIG. 381.

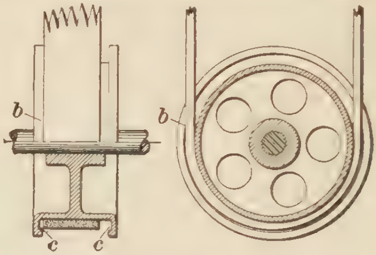


FIG. 382.

however, with the flanges grooved as at *c*, *c*, the advancing side of the belt will be guided at *b*, just as it reaches the pulley, by contact with the thick portion of the flange, and during the rest of the way will not touch the flange at all.

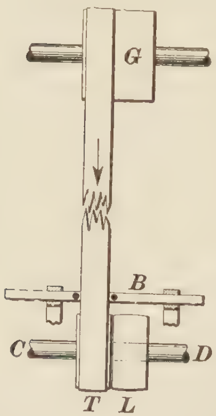


FIG. 383.

1501. In Fig. 383, is shown the arrangement for a belt shipper. *G* is the driving pulley and *T* and *L* are tight and loose pulleys on the driven shaft *C D*. *B* is the shipper, and can be moved parallel with *C D*. The acting surface, or *face*, of *G* is made straight to allow the belt to shift readily, and the faces of *T* and *L* are crowned, so that the belt will not tend to run off.

1502. The Climbing of Belts.—In Fig. 384, suppose the shafts *a*, *a* to be parallel, and the pulley *A* to be cone-shaped. The right-hand side of the belt will be pulled ahead more rapidly than the left-hand side,

because of the greater diameter and consequently greater speed of that part of the pulley. The belt will, therefore, leave its normal line at *c*, and climb to the "high" side of the pulley. This tendency is taken advantage of by crowning pulleys in the middle. Each side of the belt then tends to move towards the middle of the pulley; that is, the tendency is for the belt to stay on the pulley.

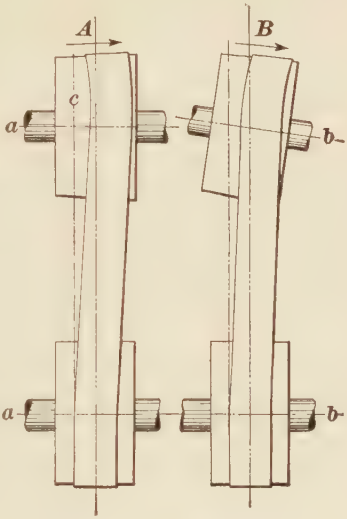


FIG. 384.

Suppose, on the other hand, the shafts *b*, *b* not to be parallel, the right-hand ends being nearer together. The belt will in that case pass spirally on the pulley *B* towards the "low" side.

1503. Belt Fastenings.—There are many good methods of fastening the ends of belts together, but lacing is generally used, as it is flexible like the belt itself, and runs noiselessly over the pulleys. The ends to be laced should be cut squarely across and the holes in each end for the lacings should be exactly opposite each other when the ends are brought together. Very narrow belts, or belts having only a small amount of power to transmit, usually have only one row of holes punched in each end, as in Fig. 385. *A* is the outside of the belt, and *B* the side running next the pulley. To lace, the lacing should be drawn half way through one of the middle holes, from the under side, as at *1*. The upper end should then be passed through *2*, under the belt and up

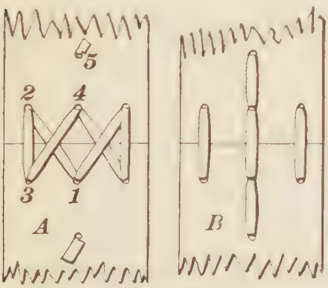


FIG. 385.

through *3*, back again through *2* and *3*, through *4* and up through *5*, where an incision is made in one side of the lacing, forming a barb that will prevent the end from pulling through. The other side of the belt is laced with the other end, it first passing up through *4*. Unless the belt is very narrow, the lacing of both sides should be carried on at once.

1504. Fig. 386 shows a method of lacing where double lace holes are used, *B* being the side to run next the pulley. The lacing for the left side is begun at *1*, and continues through *2*, *3*, *4*, *5*, *6*, *7*, *4*, *5*, etc. A 6-inch belt should have seven holes, four in the row nearest the end, and a 10-inch belt, nine holes. The edge of the holes should not be nearer than $\frac{3}{4}$ of an inch from the sides; and the holes should not be nearer than $\frac{7}{8}$ of an inch from the ends of the belt. The second row should be at least $1\frac{3}{4}$ inches from the end.

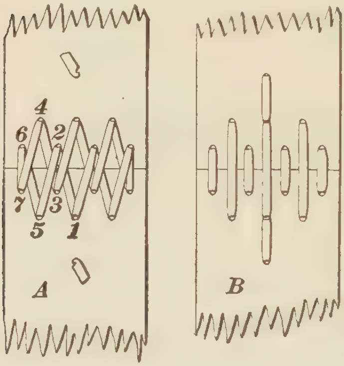


Fig. 386.

Another method is to begin the lacing at one side instead of in the middle. This method will give the rows of lacing on the under side of the belt the same thickness all the way across.

BELTS TO CONNECT NON-PARALLEL SHAFTS—GUIDE PULLEYS.

1505. It very frequently happens that one shaft must drive another at an angle with it. Sometimes this involves the use of guide pulleys, and occasionally guide pulleys must be used to connect parallel shafts, where the shafts are near together, or there is some obstruction in the way. In all these cases, however, we have the general principle that *the point at which the center of the belt is delivered from each pulley must lie in the middle plane of the other pulley.*

The middle plane of a pulley, it will be understood, is a plane through the center of the pulley, perpendicular to its axis.

Unless the shafting is to turn backwards at times, it is immaterial in what direction a belt *leaves* a pulley; but it must always be *delivered* into the plane of the pulley towards which it is running. If it is necessary for a belt to run backwards as well as forwards, it must *leave* in the plane of the pulley, also. This principle applies to the guide pulleys as well as to the main pulleys.

1506. Shafts at Right Angles.—One of the most frequent cases met with is that of two shafts at right angles, but not intersecting, and the common method of connecting them is by means of a **quarter-turn belt,** shown in Fig. 387. Here, D is the driver, revolving in the direction shown, and F is the follower. The point at which the belt is delivered from the pulley D lies in the middle plane of the pulley F, that is, in the line b b; also, the point at which the belt is delivered from the pulley F lies in the middle plane of the pulley D, or in the line a a. Thus arranged, the pulleys cannot run backwards, because d, the point of delivery of F, is not in the middle plane a a, and e is not in the middle plane b b.

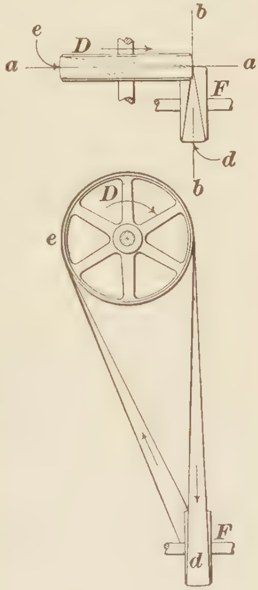


FIG. 387.

1507. The following simple method of locating the pulleys for a quarter-turn belt may be used in practice: Let D, Fig. 387, be the driving pulley, and F the follower or driven pulley, which drives a machine. Locate the pulley D and the machine so that the pulleys D and F will be as near the correct position as can be judged by the eye. Using a plumb-bob, drop a plumb-line from the center of the right-hand side of the pulley D, and move the machine until the center of the *back* side of the pulley F touches the

plumb-line. In case it should not be convenient to move the machine, shift the pulleys instead. If it be desired to run the belt in the opposite direction to that indicated by the arrow in Fig. 387, shift the machine carrying F to the left until the center of the *front* side of the pulley F touches a plumb line dropped from the center of the left-hand side of the pulley D; that is, from the point e.

1508. There is the objection to a quarter-turn belt that, when the angle at which the belt is drawn off the pulleys is large, the belt is strained, especially at the edges, and it does not hug the pulleys well. Small pulleys placed quite a distance apart, with narrow belts, give the best results, from which it follows that quarter-turn belts, like the foregoing, are not well suited to transmit much power. Fig. 388 shows how the arrangement can be improved by placing a guide pulley against the loose side of the belt. The driver D revolves in a left-handed direction, making $a\ b$ the driving or tight side of the belt. To determine the position of the guide pulley, select some point in the line $a\ b$, as d; draw lines $c\ d$ and $e\ d$; the middle plane of the guide pulley should then pass through the two lines. Looked at from a direction at right angles to pulley F, line $c\ d$ coincides with $a\ b$; looking at right angles to pulley D, line $e\ d$ coincides with $a\ b$.

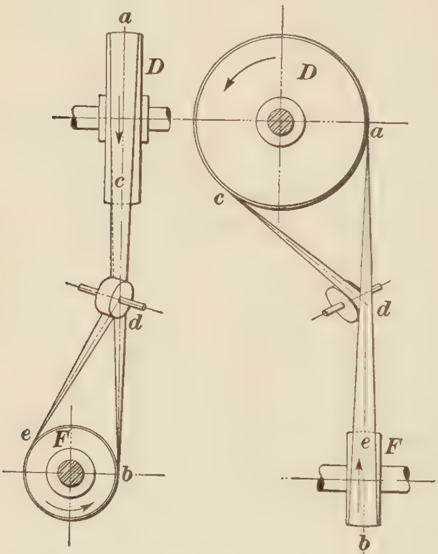


FIG. 388.

1509. A third form of belting for connecting two shafts at right angles consists of pulleys placed as in Fig. 389.

In general, it is to be preferred to the quarter-turn belts. *D* is the driver. The belt passes around the loose pulley *L*, and up again around a loose pulley on the driving shaft back

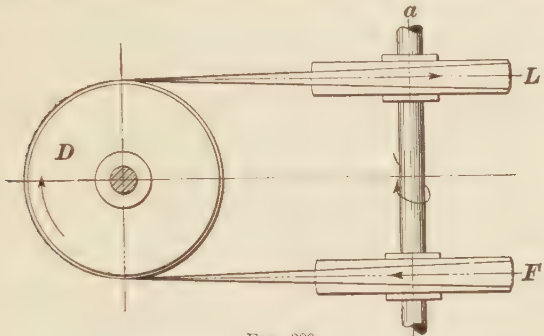


FIG. 389.

of *D*. It then goes down, around *F*, which is fast upon the shaft *a a*, and finally up again and around *D*. Since the loose pulleys revolve in a direction opposite to that of their shafts,

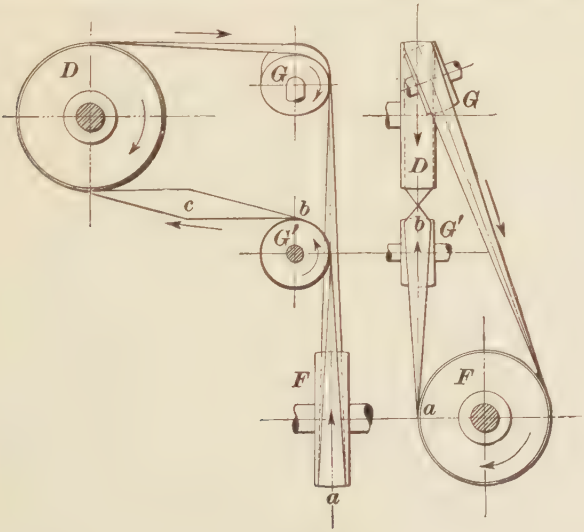


FIG. 390.

their hubs should be long. The two pulleys on each shaft must be of the same size. It is evident that either *F* or *D* can be the driver, and run in either direction.

It is to be observed that while a quarter-turn belt can be used with the shafts at an angle other than a right angle, the last arrangement cannot.

1510. In Fig. 390 is shown a method of connecting the shafts when it is not possible to put the follower *F* directly under *D*. The guide pulleys *G*, *G'* must be so placed that the belt will lead correctly from the point *a* into the middle plane of the guide pulley *G'*, from *b* into the middle plane of *D*, and so on around. By twisting the belt at *c*, the same side will come in contact with all the pulleys; this is a desirable arrangement.

1511. We now come to a case that is different from the preceding, in that the shafts *A* and *B*, Fig. 391, would

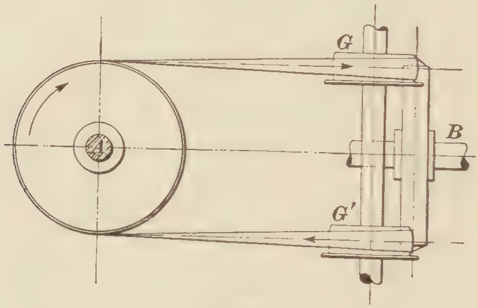


FIG. 391.

intersect, if long enough, as, for example, where shafting running on two sides of a room is to be connected. Guide pulleys, like those in the figure, termed **mule pulleys,** are used. As their planes are horizontal, means must be provided to prevent the belt from running off at the bottom. Sometimes this is done by simply crowning the pulley, and sometimes by putting flanges on the lower sides.

1512. Other Examples of Belt Transmission.—Guide pulleys are sometimes used to lengthen the belt between two shafts which are too close together to be

connected directly, or it may be that it is not possible to get two pulleys in the same plane. Fig. 392 shows an arrangement of this kind. The diameter of the guide pulleys should equal the distance between the planes of *D* and *F*. With the guide pulleys arranged as shown, the belt will run in both directions. It is more convenient, however, to place them on one shaft. In that case their axes would be on the line *O O'*. *G'* would have to be in the line *C K*, and *G* in the line *A B*. Then the belt would be delivered from *D* into the middle plane of *G'*, and from *G* into the middle plane of *F*. The belt would run in only one direction, however.

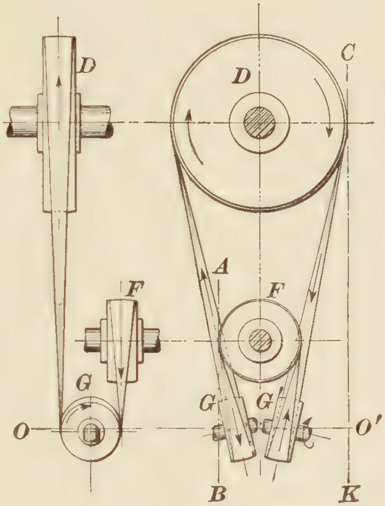


FIG. 392.

1513. A device for connecting two horizontal shafts making an angle with each other is given in Fig. 393. It can be used where a quarter-turn belt would not work successfully. The guide pulleys turn in the same direction, which is a convenience, because they can then be mounted on one shaft, turning in bearings at the ends, and the belt will run in either direction.

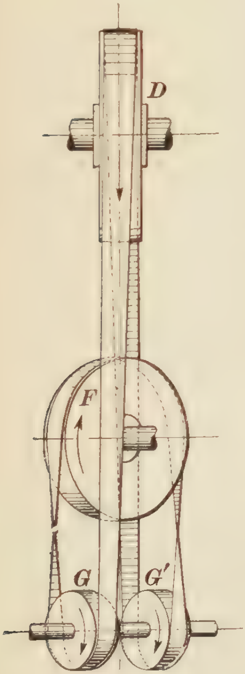


FIG. 393.

WHEELS IN TRAINS.

1514. The principles relating to the velocity ratio of pulleys connected by belting apply to any series of wheels arranged in a train. Where gears are used, however, the relative proportions of the wheels may be stated in terms of the numbers of teeth, instead of in terms of their diameters, if desired. Moreover, in any one train, it is only necessary that the proportions of each pair of wheels be stated in the same terms; different pairs may be given in different terms.

For example, take a train of four axes or shafts and six wheels, as in Fig. 394. Four of the wheels are gear-wheels,

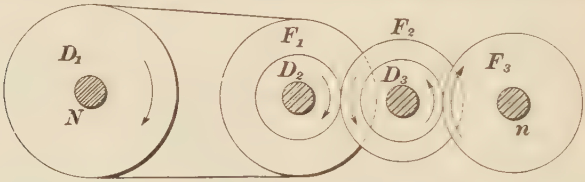


FIG. 394.

represented by their pitch lines, and two are pulleys connected by a belt. Let D_1, D_2, etc., denote the drivers, and F_1, F_2, etc., the followers, N the number of revolutions of D_1, and n the number of revolutions of F_3.

Suppose D_1 to be 40 inches in diameter and F_1 35 inches. D_2 to have 54 teeth, and F_2 60 teeth; D_3 to be 1 foot in diameter, and F_3 2 feet. What is the speed of F_3 if $N = 100$?

Placing the product of the drivers × the speed of the first shaft = the product of the followers × speed of the last shaft, $100 \times 40 \times 54 \times 1 = n \times 35 \times 60 \times 2$. Hence,

$$n = \frac{100 \times 40 \times 54 \times 1}{35 \times 60 \times 2} = 51\tfrac{3}{7} \text{ rev. per min.} \quad \text{Ans.}$$

It is evident that the arrangement of the drivers and followers is indifferent, and that they may be interchanged among themselves. It should be noticed, however, that if the diameter of one driver be given in inches, the diameter of its follower must be given in inches, also, etc.

1515. Direction of Rotation.—Axes connected by gear wheels rotate in opposite directions, as though con-

nected by a crossed belt. Hence, *in a train consisting solely of gear wheels, if the number of axes be odd, the first and last wheels will revolve in the same direction; if the number be even, they will revolve in the opposite direction.*

It is evident that a pinion working in an internal gear is an exception to this rule. It will turn in the same direction as the gear.

1516. Idle Wheels.—In the train, in Fig. 394, shaft No. 3 carried two gears, one being driven by gear D_2 and the other driving F_3. Sometimes, however, only one intermediate gear is used, serving both as a driver and a follower. Such a wheel is called an **idle wheel,** or *idler, and while it affects the relative direction of rotation of the wheels it is placed between, it does not affect their velocity ratio.*

The following examples will illustrate this, as well as show a few ways in which idle wheels are used. One method of arranging the change gears on the end of an engine lathe for changing the speed of the lead screw, which is used to feed the tool in screw cutting, is shown in Fig. 395. D_1, the driver, receives motion from the lathe spindle, and F_2, the follower, is on the lead screw. The middle wheel, which is an idle wheel, acts both as a driver and follower, or as D_2 and F_1.

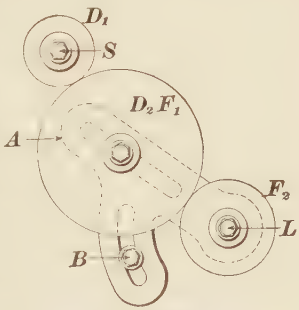


FIG. 395.

Letting N_1, represent the number of revolutions of D_1 and n_2 of F_2, we have, from formula **137,** $N_1 \times D_1 \times D_2 = n_2 \times F_1 \times F_2$. But, as D_2 and F_1 represent the same wheel, they have the same value, and $n_2 = \frac{N_1 D_1}{F_2}$. That is, the speed of F_2 is exactly the same as though no idle wheel was used.

1517. To change the speed of the lead screw, a different size wheel is put on in place of D_1, or F_2, or both. The idle wheel turns on a stud clamped in the slot in the arm A.

This stud can be moved in the slot to accommodate the different sizes of wheels on L, and to bring the wheel in contact with the gear on S, the arm is swung about L until in the right position, when it is clamped to the frame by the bolt B.

The way in which an idle wheel changes the direction of rotation is well shown in Fig. 396, which is a reversing mechanism, sometimes placed in the head stock of a lathe for reversing the feed. Here the idler I is in contact with D and F, making three axes, an odd number, so that D and F revolve in the same direction. Wheels I and I', however, are both pinioned on the plate P, which can swing about the axis of F, and are always in contact. Moreover, as plate P swings about its center, I must necessarily remain in contact with F. If the handle H be moved to the right, the plate will be revolved to the right, by means of the pin working in the slot in the lever L. The two idlers will take the dotted positions shown, and D will drive F through both of them. The number of axes will then be even, so that D and F will revolve in the opposite direction.

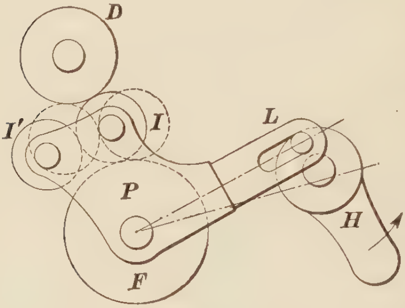


FIG. 396.

1518. Fig. 397 shows still another device in which an idle wheel is used, known as a **knee-joint.** If shaft A is fixed and shaft C is compelled by some arrangement, not shown, to move along a path, as $m\,n$, and at the same time it is desired to drive C from A, the connection can be made as

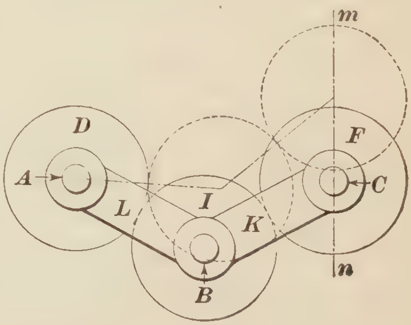


FIG. 397.

shown. Wheels D and F are fast to shafts A and C, and the idler I is supported by the links L and K, the ends of the links being loose upon shafts A, B, and C. Link L serves to keep D and I at the right distance from each other, and K serves the same purpose for I and F. The dotted lines show another position of the wheels. Among other applications of the knee-joint is one used to drive the rollers which deliver the ingot in a rolling mill to the rolls. These latter are ranged above one another in pairs, and the rollers have bearings in an iron frame, called the roll table, that is raised or lowered by hydraulic pressure, thus bringing the ingot in line with any pair of rollers. A familiar example of this train is seen in routing machines, pulleys and belts being used, however, in place of gears. The knee-joint is in reality an example of an epicyclic train, a description of which is to follow.

ENGINE LATHE TRAIN.

1519. Some of the best examples of wheels in trains are to be found in engine lathes. Fig. 398 represents the head-stock of an engine lathe, the spindle S turning in the bearings, as shown, and having a face-plate P, and a "center" on the right end for placing the work. The lead screw L, used in screw-cutting, is connected with the spindle by the train of gears on the left, which will be described later. The **back gears** F_1 and D_2, on the shaft $m\ n$, have been drawn above the spindle for convenience of illustration, instead of back of it, where they are really placed.

1520. Back Gear Train.—It is important to keep the cutting speed within limits that the tool will safely stand. For turning work of different diameters and material, therefore, the spindle must be driven at different speeds. This is accomplished by means of the cone pulley marked C, driven by a similar one on the countershaft by means of a belt, and by the back gears.

The gear F_2 is fast to the spindle and always turns with it. The cone C, however, is loose on the spindle, but can be made to turn with it by means of a lug, or catch, operated

by a nut under the rim of F_2. When the nut is moved out from the center, the lug engages with a slot on the large end of the cone. The cone will then revolve with the spindle, and as many changes of speed may be had as there

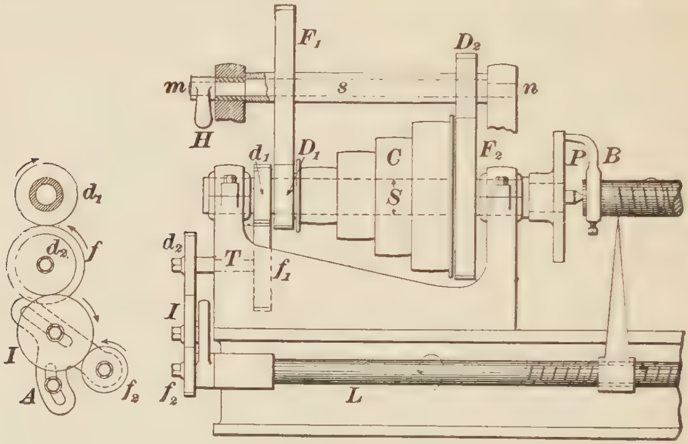


FIG. 398.

are pulleys on the cone. As ordinarily constructed, however, the cone alone does not give a range of speed great enough to include all classes of work, nor is the belt power sufficient for the larger work and heavier cuts. It makes the lathe more compact and satisfactory to construct the cone for the higher speeds and lighter work, and to obtain the speeds for the heavier work by means of back gears. Referring to the figure, it will be seen that the back gears are connected by the sleeve s, and so turn together, F_2 meshing with D_2, and D_1, which is loose upon the spindle but fastened to the cone, with F_1. To get the slower speeds, the nut mentioned before is moved in towards the center of F_2, disengaging the gear from the cone, which latter is now free to turn on the spindle. Hence, if the back gears are in mesh with the gears on the spindle, the belt will drive the spindle at a slower speed through the cone and the train $D_1 F_1 D_2 F_2$.

The back gears cannot remain in gear when the cone and gear F_2 are connected; otherwise, the lathe would not start,

or teeth would be broken out of the wheels. To provide for throwing them in and out of gear, as required, the rod on which the back gears and sleeve s revolve is provided with eccentric ends at m and n, fitting in bearings in the frame. By turning the rod part way around by means of the handle H the gears can be thrown either in or out of gear.

1521. A lathe is spoken of as running **back geared** when the back gears are in, and as being in single gear when they are out of gear. This same arrangement, or a modification of it, is used on upright drills, boring mills, milling machines, and other machine tools.

1522. Screw Cutting.—One of the chief uses of engine lathes is for screw-cutting. In Fig. 398, the screw-cutting mechanism is driven from d_1, which is fast to the spindle. This connects with the lead screw L, through f_1, d_2, the idler I, and f_2. To cut a screw-thread, the work is placed between the centers of the lathe and made to turn with the face plate by the dog B, clamped to the end of the work. The spindle runs towards the operator, or in a right-hand direction, as looked at from the outer end of the head-stock, and the carriage, tool post, and tool, all of which are here represented by the pointer, are moved by the lead screw, on shears parallel with the axis of the spindle.

When cutting a right-hand thread, the tool must move from right to left, and from left to right when cutting a left-hand thread. Hence, to cut a right-hand screw with the gearing as arranged in the figure, a left-hand lead screw must be used. To cut a left-hand screw another idle wheel should be inserted in the connecting train, arrangements for which are usually provided. To cut screws of different pitch a set of gears is always furnished, any of which may be placed on the stud T, or the end of the lead screw. The method of adjustment to accommodate the different size gears by means of the vibrating slotted arm A has already been explained.

1523. Suppose the lead screw to have six threads to the inch, or a pitch of $\frac{1}{6}$ inch, and let it be required to

cut a screw of six threads per inch. If the gearing is such that the lead screw turns once while the spindle makes one turn, it is evident that the tool will cut six threads to the inch. If it is required to cut more than six threads to the inch, however, the lead screw must turn slower than the spindle; if less than six threads, it must turn faster than the spindle.

Now, the ratio between the speed of the spindle and the speed at which the lead screw must turn for cutting a certain number of threads per inch is simply the ratio between the number of turns made by the spindle and the number of turns made by the lead screw while the tool moves along one inch. This is evidently equal to the ratio of the number of threads per inch to be cut to the number of threads per inch on the lead screw. Thus, if 10 threads per inch are to be cut, and the lead screw has six threads per inch, the spindle must turn 10 times while the lead screw turns six times; if $4\frac{1}{2}$ threads are to be cut, it must turn $4\frac{1}{2}$ times to six turns of the lead screw.

The problem, then, is to determine what size gears should be placed on the stud and lead screw to give the latter the right speed for any case. Here, as heretofore, the principle must hold that the speed of the first driver, multiplied by each driver, equals the speed of the last follower, multiplied by each follower. From this and from what has gone before we have, therefore, that *the number of threads per inch of the screw to be cut, multiplied by the number of teeth of each driver, equals the number of threads per inch of the lead screw, multiplied by the number of teeth of each follower.*

It is a simple matter, therefore, to find the ratio between the gear on the stud and the gear on the lead screw, and from that to determine what gears will be suitable.

1524. The process to be followed will be made clear by the following:

Example.—Let the number of teeth in the different wheels in Fig. 398 be as follows: d_1, 30; f_1, 60; I, idle wheel. Assuming L to have six threads per inch, what change gears should be used to cut 4 threads per inch?

SOLUTION.—By formula **137**, letting N and n be the number of revolutions of the spindle and lead screw, respectively, we have $N \times d_1 \times d_2 = n \times f_1 \times f_2$, or $4 \times 30 \times d_2 = 6 \times 60 \times f_2$. Hence, $d_2 = \frac{6 \times 60 \times f_2}{4 \times 30} = 3 f_2$.

That is, the number of teeth of the gear on the stud ($= d_2$) must equal the number of teeth of the gear on the lead screw, multiplied by three. In other words, the gear on the stud must have three times as many teeth as the gear on the lead screw.

In like manner we have

$$\text{for 5 threads, } d_2 = \frac{12}{5} f_2;$$
$$\text{for 6 threads, } d_2 = 2 f_2;$$
$$\text{for 7 threads, } d_2 = \frac{12}{7} f_2;$$

etc., etc., etc.

For four threads we might use wheels of 84 and 28 teeth; for 5 threads 84 and 35, etc. In this way, a table might be calculated and arranged in columns, as below:

Threads per Inch.	Gear on Stud.	Gear on Lead Screw.
4	84	28
5	84	35
6	84	42
7	84	49
etc.	etc.	etc.

Such reference tables are always provided with lathes, and the gears are generally so chosen that the smallest number possible will have to be furnished with the lathe to cut the range of threads desired. Thus, in the above table, the gears were chosen so that one with 84 teeth would serve for cutting several different threads.

1525. Many lathes adapted for screw-cutting are not provided with the stud T, the gear d_2 being keyed directly to the end of the spindle S.

In such a case, if it is desired to find whether the lathe will cut a certain thread not marked on the plate which is usually attached to the head-stock, write the number of threads to be cut as the numerator of a fraction and the number of threads on the lead screw as the denominator; multiply both numerator and denominator by some number, and ascertain whether any of the gears in stock have the number of teeth corresponding to the results obtained; if not, multiply again by some other number, etc. Thus, suppose the lead screw has 8 threads per inch and it is desired to cut a thread which shall have $7\frac{1}{2}$ threads per inch. Forming the fraction gives $\frac{7\frac{1}{2}}{8}$. Multiplying both numerator and denominator by 6 gives $\frac{45}{48}$. Should any of the gears have 45 and 48 teeth, the 45-tooth gear should be put on the lead screw and the 48-tooth gear on the spindle.

1526. Again, when the lathe has a stud, the gears d_1 and f_1 are never changed, and the required change gears may be found by multiplying the fraction formed as directed above by a second fraction whose numerator shall be the number of teeth in d_1 and denominator the number of teeth in f_1, and then proceeding as before.

Thus, suppose it is desired to cut a thread having 19 threads per inch, the lead screw having 8 threads; the gear d_1, 24 teeth, and the stud pinion f_1, 48 teeth. Then, $\frac{19}{8} \times \frac{24}{48} = \frac{19}{16}$; and $\frac{19}{16} \times \frac{4}{4} = \frac{76}{64}$; hence, a gear having 76 teeth placed upon the lead screw, and one having 64 teeth placed on the stud, will cut a screw having 19 threads per inch.

REVERSING MECHANISMS.

1527. It is frequently required to design machinery, parts of which must have a reciprocating motion alternately in opposite directions. The various crank motions are sometimes employed for this purpose, but in many cases their

use is inadmissible, as they produce a variable motion, and for other reasons are inconvenient. It is also often necessary that machinery be constructed to run in either direction, at will. All such cases require the use of some reversing mechanism other than those previously described. A few additional examples will now be explained.

1528. Mangle Gearing.—Figs. 399 and 400 show the principle of mangle gearing, so named from its use in

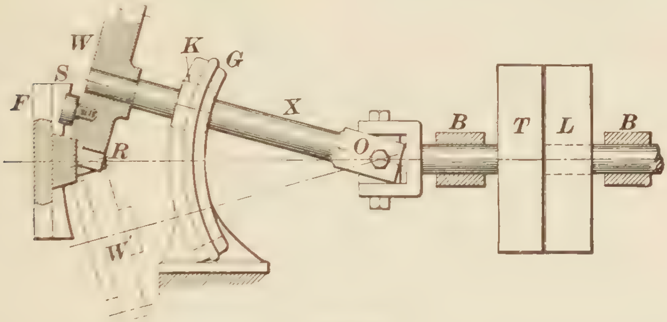


FIG. 399.

mangles for pressing clothes. Its principal use, at the present time, is for actuating the tables of printing presses. In Fig. 399 is a rack *R*, shown in cross-section, and so constructed that the gear-wheel *W* can run in mesh with it, either on top in position *W*, or underneath, as at *W'*. The rack is fastened to a frame *F*, also in section, attached directly to the table of the press. The object is to drive the table back and forth in a direction to and from the reader. The method will be understood by reference to Fig. 400, which is the rack and gear of Fig. 399, looked at from the left. As the frame *F* would obstruct the view of the other parts, it is omitted. A longitudinal section of the rack is shown.

Suppose the gear *W* to be on top of the rack, and to turn uniformly in the direction indicated. The rack will be driven to the right with a velocity equal to that of the pitch circle of the gear, until the roll *A* comes under the gear. One tooth of the gear is here omitted, leaving a large space which is rounded to fit the roll. From this point the gear

will gradually drop down in the line *m n*. At position *2* it will have made a quarter-turn, and the rack will have moved a distance r (= radius of the gear) to the right. At position *3* the rack will be at its former place, and further rotation of the gear will drive it to the left at a uniform speed. The motion of the rack is harmonic during the

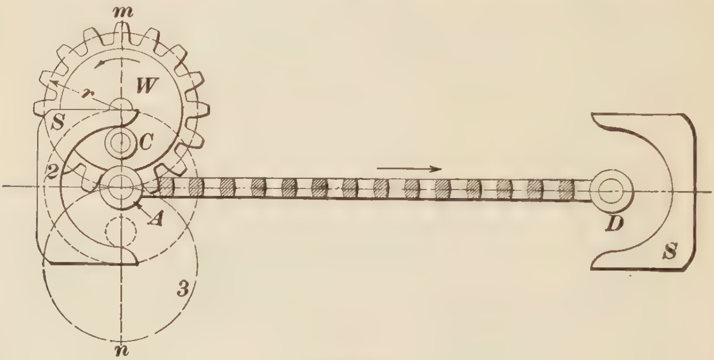


FIG. 400.

reversal. To place it under full control of the gear at the points of reversal, "shoes" *S*, *S* are fastened securely to the frame *F*. The inner curves of these are concentric with *A* and *D*, respectively, and serve as surfaces for the roll *C* on the gear to roll upon.

It is evident that the length *A D* of the rack must be equal to the circumference of the gear, or some multiple of the circumference, so that the wide space will come in contact with the rolls at each end. If, however, two spaces are cut diametrically opposite to each other, length *A D* may be some multiple of the half circumference. The total length of the stroke of the table is *A D*, plus the diameter of gear.

1529. Referring again to Fig. 399, the driving mechanism consists of tight and loose pulleys *T* and *L* on the shaft running in bearings *B*, *B*. *O* is a universal joint which allows the gear to rise and fall. *K* is a square block loose on the gear shaft *X*. It slides in a slotted guide *G*. and constrains the gear to move in a vertical straight line at the points of reversal.

The advantages of this gearing are that, neglecting the irregular motion of the universal joints, a uniform reciprocating motion of any length of stroke may be given the table with a gradual reversal. The disadvantage is that, where once constrained, the stroke cannot be adjusted.

1530. Clutch Gearing. — A reversing mechanism often used, especially when the reversal must take place automatically, is shown in Fig. 401. It consists of three

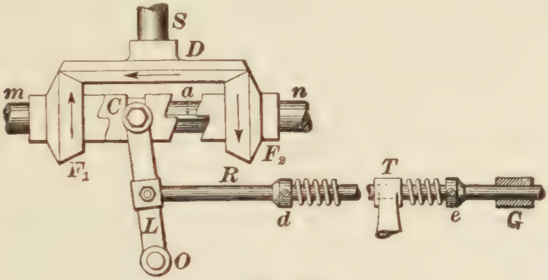


FIG. 401.

bevel gears, of which D is the driver, fast on the driving shaft S. F_1 and F_2 are continually in contact with D, and are loose upon shaft $m\,n$. A sleeve C can move endwise upon $m\,n$, but is compelled to turn with it by a key a provided in the shaft. When in the position shown, the notches clutch with F_1, and $m\,n$ turns; when in mid-position, $m\,n$ will not turn, and when thrown to the right, the direction of rotation will be reversed.

1531. If the reversal is to be automatic, some provision must be made to insure that after C has been pushed out of contact on one side, it will be thrown in contact with the other gear. One way of doing this is shown in the figure. The lever L is pivoted at O, and at the other end is forked to embrace C, a roller on each prong running in the groove shown. On the rod R, which is pivoted to L at one end and slides in a guide G at the other end, are two collars d and e, held in place by set screws. Helical springs are also placed on R against the inside of the collars. Now, suppose the tappet T, which is free to slide on R, be given a motion to

the right through mechanism connected with F_1, but not shown in the cut. When it reaches the spring, it will compress it until the pressure of the spring on e is sufficient to overcome the resistance of the clutch. Further movement of T will move C to the right until free of F_1, when the energy stored in the spring will suddenly throw C in contact with F_2. The time at which reversal occurs can be adjusted by changing the position of d and e.

1532. Sometimes it is desirable to have a slow motion in one direction with a "quick return." Figs. 402 and 403 show two methods that may be used for this purpose. In the first, the driver D is made cup shaped so as to allow a smaller driver d to be placed inside. For the slow motion we have d driving F_2, and, for the quick return, D driving F_1. In Fig. 403, S is the driving shaft, giving a slow and powerful motion through the worm gearing, shown clearly in the end view. The quick-return motion is through the gears D_1, F_1, D_2, and F_2.

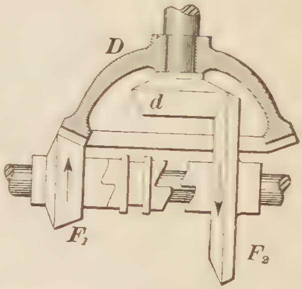


FIG. 402.

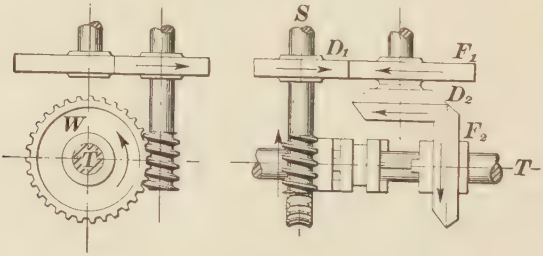


FIG. 403.

1533. It is evident that these mechanisms are not suitable for quick-running or heavy machinery. For such use, two belts are generally employed in place of the gears, one open and one crossed. Often these belts are made to shift alternately from a tight to a loose pulley, while in other cases they are arranged to drive two pulleys on the same

shaft in opposite directions, either of the pulleys being thrown in or out by means of a friction clutch, as, for example, in lathe countershafts.

1534. Shifting-Belt Mechanism.—Of the former class, Fig. 404 affords an illustration, as applied to a planer operating on metal. *H* is a section of the table, which is driven

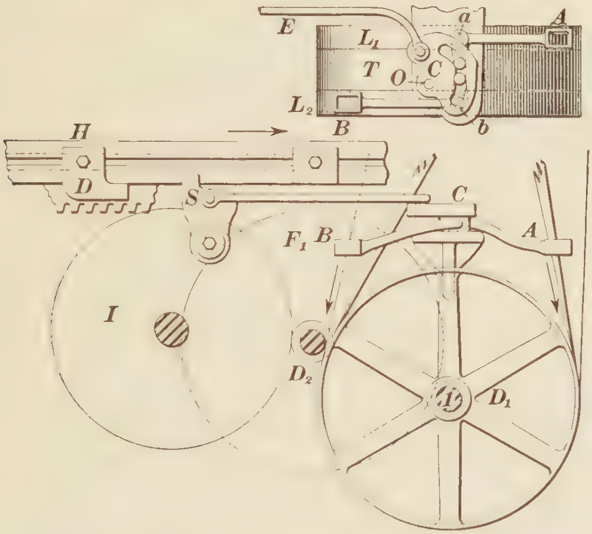


FIG. 404.

forwards and backwards by the rack and gearing shown. The work to be planed is clamped to the table, and during the forward, or cutting, stroke a stationary tool removes the metal. As no cutting takes place on the return stroke, the table is made to run back from two to five times as fast as forward, the latter speed being about 18 feet per minute for work on cast iron. There are three pulleys on shaft *1*. *T* is the tight or driving pulley, and L_1 and L_2 are both loose (see top view). There are two belt shifters *A* and *B*, in the shape of bell-cranks, pivoted at *a* and *b*, respectively. A crossed belt, moving in the direction of the arrow, runs from a small pulley on the countershaft, and is guided by the shifter *A*. This belt drives the table during the cutting

stroke. The other belt is "open," runs over a large pulley on the countershaft, and is guided by the shifter B.

The short arms of the shifters carry small rollers working in a slot in a cam plate C, which is pivoted at O. Each end of this slot is concentric about O, but one end has a greater radius than the other. As shown, the two shifters are in mid-position, both belts being on the loose pulleys. Suppose, however, the rod E be pulled to the left. Shifter B will not move, because its roller will continue to be in the same end of the slot, which is concentric about O; but the roller on A will be pulled to the left by passing from the end of the slot, which is of larger radius, to the end of the smaller radius. The crossed belt will, therefore, be shifted to the tight pulley, which will cause the table to run forward until the dog D strikes the rocker S, throwing the cam in the other direction. The effect of this will be to first shift the crossed belt from the tight to the loose pulley, and then to shift the open belt to the tight pulley; one motion follows the other, and thus decreases the wear and tear of the belts. The length of the stroke of the table can be varied by changing the position of the dogs, which are bolted to a T slot on the edge of the table.

The table is driven as follows: The belt pulleys drive gear D_1, which is keyed to the same shaft; D_1 drives F_1, which in turn drives D_2, keyed to the same shaft. D_2 drives gear I, which drives the table by means of a rack underneath it. The circumferential speeds of I and D_2 are evidently the same, and the speed of the table is the same as the circumferential speed of the gear D_2. The velocity ratio of the gearing is generally expressed in terms of the number of feet traveled by the belt to one foot passed over by the table.

1535. Suppose the belt pulleys to be 24 inches in diameter, and to make N revolutions while the table travels one foot. Let the diameter of D_1 be 3 inches; of F_1, 26 inches, and of D_2, 4 inches. The circumference of D_2 is $12\frac{1}{2}$ inches, nearly, so that for every foot traveled by the table, D_2

(and F_1) will turn $\frac{12}{12\frac{1}{2}} = \frac{24}{25}$ times. To find the number of turns made by the belt pulleys for each foot passed over by the table, we have, therefore,

$$N \times 3 = \frac{24}{25} \times 26, \text{ or}$$

$$N = \frac{24 \times 26}{75} = 8.32.$$

This multiplied by the circumference of the pulley = 8.32 × 2 × 3.1416 = 52¼ + feet. That is, the planer is geared to run 52¼ to 1.

EXAMPLES FOR PRACTICE.

1. In a wheel train, a pulley A drives a pulley B by open belt; B is on the same shaft with a gear C, which drives a gear D; a gear E is on the same shaft with D and meshes with a gear F. Suppose A to be 30″ in diameter and B 26″, and let C have 90 teeth; D, 80 teeth; E, 70 teeth, and F, 60 teeth. How many revolutions will F make in five minutes, if A runs at 45 revolutions per minute?

Ans. 340.74 revolutions.

2. An engine lathe belt is on the third speed, the diameters of the steps upon which the belt is running being 9″ on the countershaft and 4″ on the lathe spindle. If the back gears are "in," how many turns will the countershaft make for one of the lathe spindle, supposing gears F_1 and F_2 (Fig. 398) to have 64 teeth each, and D_1 and D_2, 24 teeth each?

Ans. 3.16 revolutions.

3. In Fig. 398, suppose the stud T to make 3 turns while the lathe spindle makes 4. What change gears could be used to cut 10 threads per inch, the lead screw having 6 threads per inch?

Ans. The gears should be in the ratio of 4 : 5, the latter being the lead screw gear.

4. In Fig. 404, let the diameter of the pulleys be 30″; of D_1, 4″; of D_2, 4″, and of F_1, 22″. What is the ratio of belt to cutting speed?

Ans. 41.4 + : 1.

DIFFERENTIAL GEARING.

1536. Heretofore, in the discussion of wheel trains, it has been assumed that the bearings of the various wheels remained fixed. Occasionally, however, cases are met with in practice where the wheels not only turn about their own axes, but in which one or more of them revolves bodily about

some other axis, thus having a compound motion of rotation and translation. Such combinations are known as **differential gearing** or **epicyclic trains.** Their action is somewhat confusing at first, but can be made to appear simple by applying the principle of successive movements.

1537. The Two-Wheel Train.—I.—In Fig. 405 are shown two gears D and F united by an arm A. Suppose the number of teeth in D to be represented by m, and the number in F, by n; then, when the arm A is fixed, and D is revolved once about its axis right-handed, F will be revolved left-handed about its axis $\frac{m}{n}$ times. Denoting right-hand rotation by + and left-hand rotation by −, gear D turns + 1 times and gear F, $-\frac{m}{n}$ times. This is the ordinary case of transmission of motion by two gears.

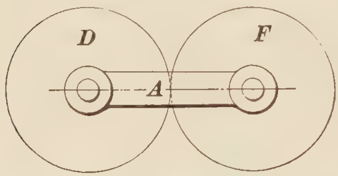


FIG. 405.

1538. II.—Suppose, however, that gear D is fixed in position so that it cannot turn, and the arm A is given a + rotation about the axis of D, the gear F then partakes of two rotations, + 1 about the axis of D and $+\frac{m}{n}$ about its own axis. In order to see this more clearly, imagine the two gears to be replaced by two friction wheels whose diameters are in ratio $\frac{m}{n}$; then, it is perfectly clear that, in order to rotate the arm, the wheel F must roll on D in the *same* direction, and that the number of times it turns on its axis is $\frac{m}{n}$. The only difference in the motions, in so far as the rotation of F on its own axis is concerned in these two cases, is that it has a negative rotation in the first case and a positive one in the second case. If the two gears are equal, one revolution of the arm will cause F to turn once on its own axis. This may also be considered in the following manner:

(*a*) Let the wheels be locked, or wedged together, so that neither can turn relatively to the other, and let the whole combination, *D*, *F*, and *A*, be turned as one body about the axis of *D* once R. H. (right-handed). The arm *A* has now been turned around once R. H., just as was desired; but, in doing so, *D* has been turned once R. H., when, according to the conditions, it should have remained stationary.

(*b*) Hence, the next step is to unlock the wheels, hold the arm stationary and turn *D* back one turn L. H. (left-handed) to where it should be. This will cause *F* to turn $\frac{m}{n}$ times R. H. After that is done, each part of the combination will have been through just the same relative motion that it would have had if the conditions had been carried out directly.

The results of the two steps upon *D*, *F*, and *A* can be tabulated thus, + indicating R. H. rotation and − indicating L. H. rotation:

	D	F	A
(*a*) Wheels locked	$+1$	$+1$	$+1$
(*b*) Arm stationary	-1	$+\frac{m}{n}$	0
(*c*)	0	$1+\frac{m}{n}$	$+1$

The algebraic sum of these in the third horizontal row gives the total motion of each part. It will be seen from this that if *D* and *F* have the same number of teeth, *F* will make two revolutions to one of the arms, one being about its own axis, and the other about the axis of *D*.

EXAMPLE.—Referring to Fig. 405, suppose *D* has 50 teeth, *F*, 20 teeth, and *A* to be turned 10 times L. H. How many turns will *F* make, and in what direction?

SOLUTION.—	D	F	A
Wheels locked	-10	-10	-10
Arm stationary	$+10$	$-10 \times \frac{50}{20}$	0
	0	$-(10+25)$	-10

F makes 35 turns L. H.

1539. III.—Suppose that the shaft on which F turns were the crank-shaft of a steam engine, and that the gear D were keyed to the end of the connecting-rod, the arm A being loose on both shafts. Then, one stroke of the piston would carry D to a position diametrically opposite, and the return stroke would bring it back to its first position; in other words, D would pass entirely around F, but without turning on its own axis, because of being keyed to the connecting-rod. The result will be that F will turn twice for one revolution of the arm when the gears D and F are of the same size. Although difficult to explain in a simple manner, a little thought will convince the student of the truth of this statement, and if he can obtain a couple of gears and try the experiment the result will amply reward him for the trouble. The result may be arrived at very simply, however, by means of the method of analysis used above. Suppose the gears to be locked and the whole combination, including the connecting-rod, to be revolved once right-handed. Denoting the connecting-rod by C, and remembering that the connecting-rod as a whole does not revolve, we must now return it to its original position by giving it a left-hand rotation (the arm A being fixed). This causes D to turn once left-handed and F, $\frac{m}{n}$ times right-handed, assuming that D has m teeth and F has n teeth. Tabulating the results, we have

	C	D	F	A
Wheels locked	$+1$	$+1$	$+1$	$+1$
Arm stationary	-1	-1	$+\frac{m}{n}$	0
	0	0	$1+\frac{m}{n}$	$+1$

The algebraic sum shows that for one revolution of A, F turns $1+\frac{m}{n}$ times and that C and D do not turn at all, which is perfectly true, since, when D is keyed to C, it is no longer a separate link, but is a part of C. Consequently, if D and

F are of the same size, $m = n$ and F turns twice right-handed for one right-handed revolution of the arm A.

This motion was used by Watt to drive his engine, some one else having patented the crank motion. It is known as the **sun and planet motion;** the fixed gear being the sun and the revolving gear the planet.

1540. Higher-Wheel Trains.—In Fig. 406 is represented a three-wheel train, derived from the two-wheel train by inserting the idle wheel I. Here, as in previous cases, the number of turns made by the wheel F is entirely independent of the size or number of teeth in the idler.

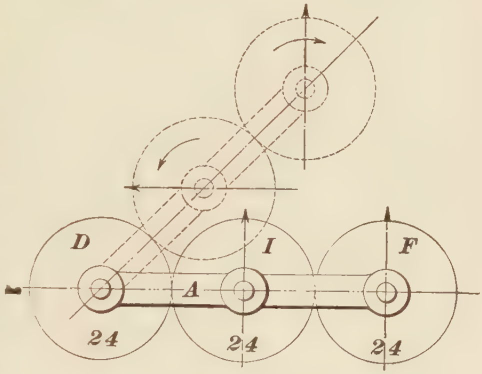


FIG. 406.

EXAMPLE.—Suppose all three wheels to have 24 teeth. If the arm makes -5 turns about the axis of D, how many turns will F make, and in which direction?

SOLUTION.—

	D	I	F	A
Wheels locked	-5	-5	-5	-5
Arm stationary	$+5$	-5	$+5$	0
	0	-10	0	-5

The method of procedure is exactly the same as with the two-wheel train, and it will be noticed that the action of I in this train is the same as that of F in the two-wheel train. F, in the three-wheel train, however, does not turn at all. The straight arrows in the figure are supposed to be fastened to the wheels, so that the action of the train can be seen by noticing the directions in which the arrows point in the dotted positions.

1541. The statement that F does not turn at all is not literally true, since it turns once on its own axis and once on the axis of D. What the statement really implies is this:

The gear F cannot impart motion to an annular gear (supposing it to mesh with one) keyed to the shaft of D; that is, if an annular gear, keyed to the axis of D, meshes with gear F, and is caused to revolve owing to the rotation of the arm A, it will receive no motion from the gear F owing to the rotation of F on its own axis. It will, in fact, be deprived of a part of the rotation it should receive from the arm of A owing to this positive rotation of F, the amount of which is represented by $\frac{m}{p}$, m being the number of teeth in each of the three gears and p the number of teeth in the annular gear. Suppose, however, that the three gears D, I, and F were to be replaced by the two gears D and F, of Fig. 405, both gears being free to turn on their axes. Then, representing the number of teeth in the annular gear by p, one + revolution of the arm will cause the annular gear (which we will call L) to rotate $1 + \dfrac{\frac{m}{n} \times n}{p} = 1 + \dfrac{m}{p}$ times. For the annular gear and gear F, both turn right-handed (D being fixed) and F turns $\frac{m}{n}$ times; the number of teeth in L which come in contact with corresponding teeth in F is $\frac{m}{n} \times n = m$; hence, L will make a part of a revolution represented by $\frac{m}{p}$ due to the turning of F on its axis, and one revolution due to the rotation of the arm A, the whole movement being represented by $1 + \frac{m}{p}$. Using our method of analysis and applying to the first case, we have, assuming all three gears attached to A to be of the same size, and remembering that an annular gear always turns in the same direction as its pinion,

	D	I	F	L	A
Wheels locked	$+1$	$+1$	$+1$	$+1$	$+1$
Arm stationary	-1	$+1$	-1	$-\frac{m}{p}$	0
	0	$+2$	0	$1-\frac{m}{p}$	$+1$

For the second case, with two gears,

	D	F	L	A
Wheels locked	$+1$	$+1$	$+1$	1
Arm stationary	-1	$+\frac{m}{n}$	$+\frac{\frac{m}{n}\times n}{p}$	0
	0	$+1+\frac{m}{n}$	$+1+\frac{m}{p}$	$+1$

It should not be imagined that the gear F imparts a backward motion to L in the first case above, for it does not; its rotation simply prevents the arm from imparting to L its entire motion, which it would do if F were keyed to the arm.

1542. A four-wheel train, one of the wheels being an annular wheel, is shown in Fig. 407. D is fixed, the other wheels all turning. A revolves about the axis of F_1. D_1 and F are on the same spindle, F rolling inside of D.

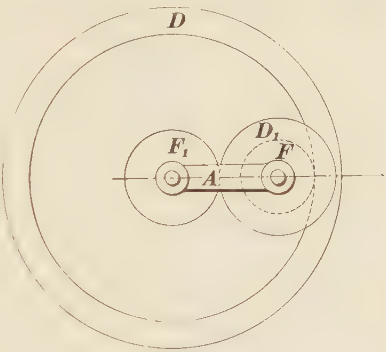


FIG. 407.

In working out a train of this kind, first consider all the wheels, including D, to be locked together and to turn with the arm, as in previous cases. Then, make the arm stationary, and turn D back to the position it should occupy.

EXAMPLE.—Let D have 100 teeth; F, 20; D_1, 45, and F_1, 35. A makes $+10$ turns. Required the number of turns made by the pinions.

SOLUTION.—	D	F and D_1	F_1	A
Wheels locked	$+10$	$+10$	$+10$	$+10$
Arm stationary	-10	-50	$+\left(\frac{45}{35}\times 50\right)$	0
	0	-40	$+74\frac{2}{7}$	$+10$

1543. Differential Bevel Train.—A **differential bevel train** is shown in Fig. 408. It consists of three miter wheels, two of which, E and C, are on the shaft S, the

third revolving on the arm A, which is also on S. It is assumed that none of the wheels can slide endways upon their shafts. This train has certain peculiar as well as very useful properties.

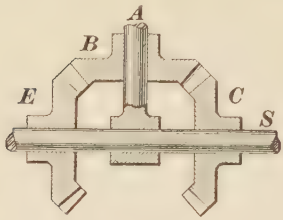


FIG. 408.

Suppose A to be held rigidly in one position, and E and C to be loose on the shaft. Then, if C be turned once one way, E will turn *once* in the *opposite* direction, the three wheels forming a simple train of gears, B being an idle wheel.

Now, suppose C to be held in one position and A to be turned once about the shaft S. E will then revolve *twice* in the *same* direction as the arm. That it will do so can be clearly seen by applying the principle of relative motions, as follows:

	A	C	E
Wheels locked	+1	+1	+1
Arm stationary	0	−1	+1
	+1	0	+2

From the above it will be seen that if we consider C the driver and E as fixed, A will revolve one-half as fast as C.

If we assume that the arm and all the wheels are free to turn, and if C is rotated one way at a uniform speed and E the opposite way at the same speed, B will revolve upon A, but A will remain still. If, however, C is rotated faster than E, the arm will move in the direction of C through one-half the angle gained; if C turns slower than E, the arm will move correspondingly the opposite way.

It will thus be seen that a bevel train is capable of a variety of combinations that can be applied in a useful way. A few practical applications of this and the other differential trains will now be given.

EXAMPLES FOR PRACTICE.

1. In Fig. 406, let D have 25 teeth; I, 30 teeth, and F, 50 teeth. If D is stationary and the arm makes 5 turns L. H., how many turns will F make, and in what direction?

Ans. $2\frac{1}{2}$ turns L. H.

2. In Fig. 407, suppose D to be 40″ in diameter; F, 10″; D_1, 14″, and F_1, 16″. If D remains stationary, how many turns will F_1 make to each turn of the arm? Ans. $4\frac{1}{2}$ turns.

3. In Fig. 408, suppose E to make 1 turn R. H. (looked at from the right) and A, two turns L. H. How many turns will C make, and in which direction? Ans. 5 turns L. H.

1544. Differential "Back-Gears."—Upright drills for metal work are sometimes provided with arrangements for increasing the range of the speeds and driving power that are different from the back-gears explained under the engine-lathe train. Fig. 409 shows one arrangement for this purpose. S is the shaft, and C the cone-pulley which is loose on the shaft. D is a casting, also loose on the shaft, having

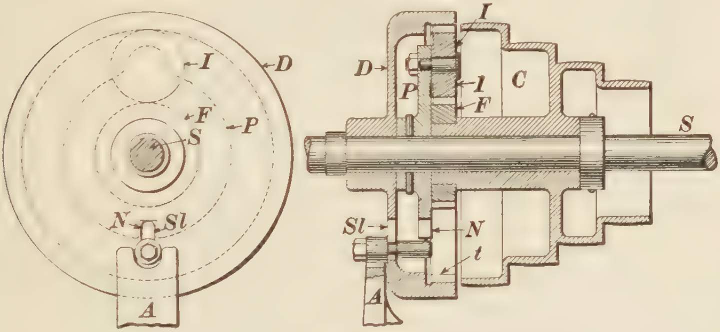


FIG. 409.

teeth on the inside, thus forming an annular gear. A plate P, carrying the small gear or pinion I, is fast to the shaft. On the left-hand end of the pulley hub is another gear F, which is fast to the hub of the cone.

The action is as follows: A pin, on which there is a collar and nut, is clamped in the slot $S\,l$ in D. The pin projects through D so that when it is placed in the inner end of the slot it will engage with a corresponding slot in the plate P. When it is lowered, however, the pin disengages with the plate, but the collar can be made to fall in a slot in the arm A, which is a part of the frame. In the former position D, P, and F are locked together, so that the shaft must turn with the cone. In the latter position D is locked with the

frame and cannot turn, while F revolves with the cone. The plate P, therefore, revolves with the shaft at a reduced speed, the arrangement being similar to that in Fig. 407.

1545. Differential Motion.—Perhaps the most important application of the differential bevel train is to be found in spinning machinery, where it is necessary to wind the partly twisted fiber, or *roving*, upon bobbins. As each successive layer is wound on the bobbin, the latter becomes correspondingly larger and must revolve at a reduced speed; otherwise, the roving, which is delivered at a constant speed, would be broken.

1546. In Fig. 410, B is the bobbin at the top of which the flyer f is suspended. The roving passes through a hole c in the center of the upper part of the flyer, then down

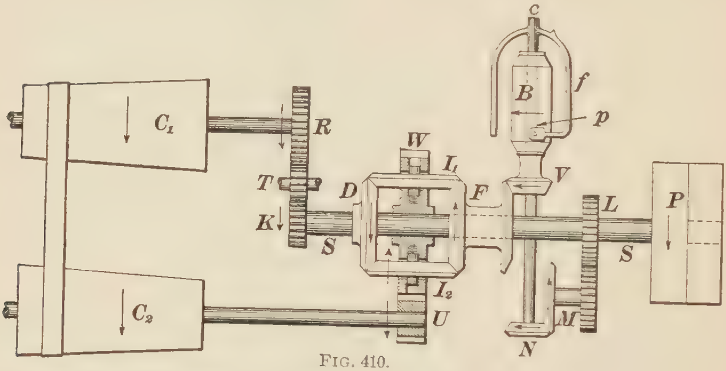


FIG. 410.

through the hollow arm f of the flyer and through a hole in the presser p to the bobbin. This gives a uniform twist to the roving. The flyer moves up and down the length of the bobbin winding on the roving in helical layers.

The machine is driven by a belt through the pulley P and the shaft S S. The first gear L on the shaft drives the flyer at a constant velocity through M and N. The next gear that is fast to the shaft is the miter wheel D. This forms part of a differential bevel train, the other wheel on the shaft being F, which is loose. To improve the running qualities, there are two intermediate wheels, I_1, I_2, instead of one, as in Fig.

408. A gear W, shown in section and loose upon the shaft, serves the purpose of the arm used in Fig. 408. Its action is exactly the same as the arm, the wheel being used simply as a means for causing it to revolve about the shaft. The object of the epicyclic train is to drive the bobbin, by means of F and V, at a *varying* velocity suited to the varying diameter of the bobbin.

This is brought about by the cone pulleys C_1 and C_2. The former is driven by the main shaft through the gears K, T, and R, and the latter drives W through gear U. Now we have D rotated uniformly in one direction, which turns F the opposite way. W has a motion opposite to that of D, the number of revolutions that it makes depending upon the position of the belt on the cones. As the belt is moved automatically to the left, W turns slower, F turns slower, and hence the bobbin turns slower as the roving is wound on. The gearing is such that the bobbin revolves in the same direction as the flyer and just enough faster to take the roving as it is delivered.

1547. Hartford Water-Wheel Governor.—Another application of the bevel train is to be found in the water-wheel governor, in Fig. 411. W_1 and W_2 are two wide-faced pulleys driven by the belt which passes over the guide pulleys P_1 and P_2. W_1 is a frustum of a cone, with its diameter at the middle, the same as that of W_2, which is an ordinary pulley. These pulleys give motion to the double bevel wheels E and C, both of which are loose upon the shaft.

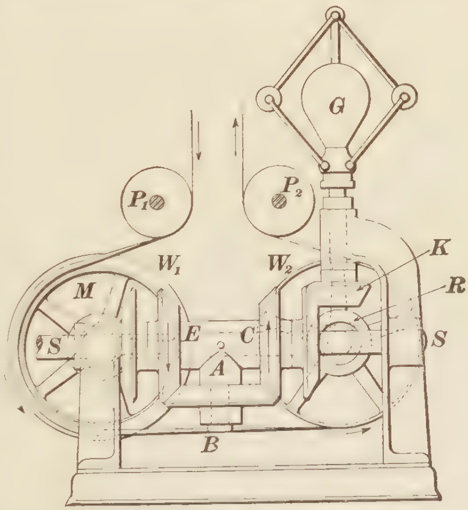


FIG. 411.

through the bevel gears M and R, respectively. The inner parts of E and C gear with the miter gear B on the arm A, the latter being keyed to the shaft. The outer part of C drives the governor G through the gear K. The work of the governor is to shift the belt on the cone by means of a fork (not shown) near pulley P_1.

The action is as follows: When the water-wheel is running at the proper speed, the belt is at the middle of the cone and E and C turn equally in opposite directions. Hence, A does not move. Should the wheel run too fast, however, the governor balls would rise, since the speed of W_2 must always be proportional to that of the water-wheel. This at once causes the belt to be shifted to the small end of the cone, increasing the speed of E. This, in turn, makes A, and hence the shaft, rotate in the direction of E, and an arrangement at the left-hand end of the shaft $S\ S$ (not shown) lowers the water-wheel gate. If the wheel should slow down, the reverse operation would take place.

GEAR WHEELS.

1548. Let two wheels with parallel axes be held in firm

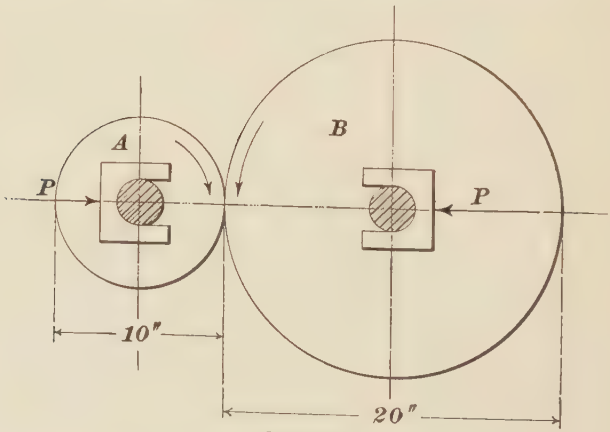


FIG. 412.

rolling contact by pressure upon their axes, as in Fig. 412.

If one be turned in either direction, and there is no slipping, the other will rotate in the opposite direction with a circumferential, or surface, velocity equal to that of the first, as though connected by a crossed belt, and the numbers of their revolutions will be inversely proportional to their diameters. Assuming wheel A to be 10″ in diameter and B 20″, B would make $\frac{10}{20} = \frac{1}{2}$ as many revolutions as A.

1549. Should slipping occur, however, B would make less than one-half as many revolutions as A, if A were the driver. To obviate slipping, suppose that pieces like a, a, Fig. 413, are fastened at equal distances on the peripheries of A and B, and that corresponding grooves like b, b are cut. Then, the projections, or teeth, on one wheel will run be-

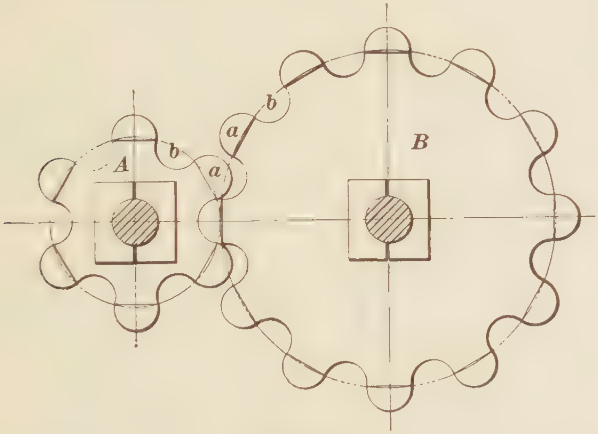


FIG. 413.

tween the teeth on the other, and B will necessarily revolve one-half as often as A. It is important to notice, however, that although the *number of revolutions* of the wheels in the latter case will be in the ratio of 2 to 1, the speed from *tooth to tooth* might vary somewhat from this ratio, unless the teeth were of a shape that would give a constant velocity ratio. This would result in an uneven motion that would be undesirable, even though the variation was very slight.

1550. The object, then, in designing the teeth of gear wheels should be to so shape them that the motion transmitted will be exactly the same as with a corresponding pair of wheels, or cylinders, without teeth, and running in contact without slipping.

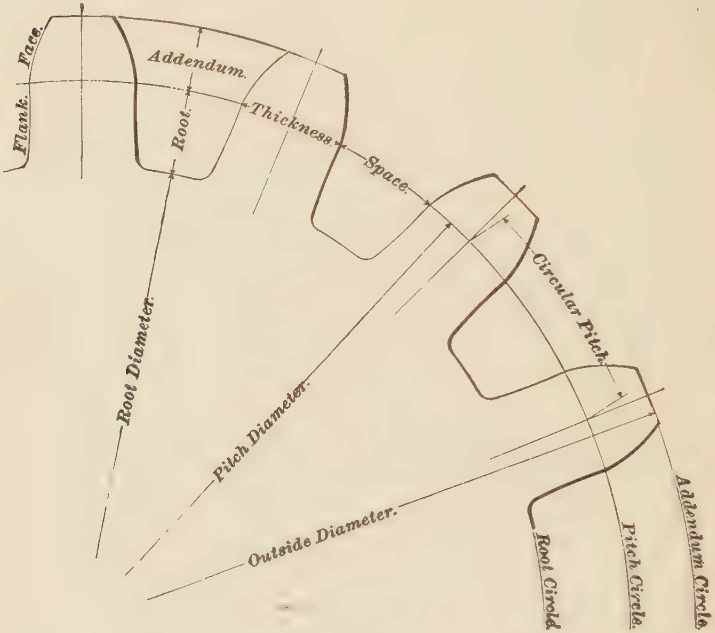


FIG. 414.

Such cylinders are called **pitch cylinders**, and are always represented on the drawing of a gear-wheel by a line called the **pitch circle**. (See Fig. 414.) The pitch circle is also called the **pitch line.**

1551. The diameter of the pitch circle is called the **pitch diameter.** When the word "diameter" is applied to gears, it is always understood to mean the **pitch diameter,** unless otherwise specially stated, as "outside diameter" or "diameter at the root."

1552. Circular and Diametral Pitch.—The distance from a point on one tooth to a corresponding point on the next tooth, *measured along the pitch circle*, is the **circular pitch.** It is obtained by dividing the circumference (pitch circle) by the number of teeth, and is used in laying out the teeth of large gears, and also when calculating their strength.

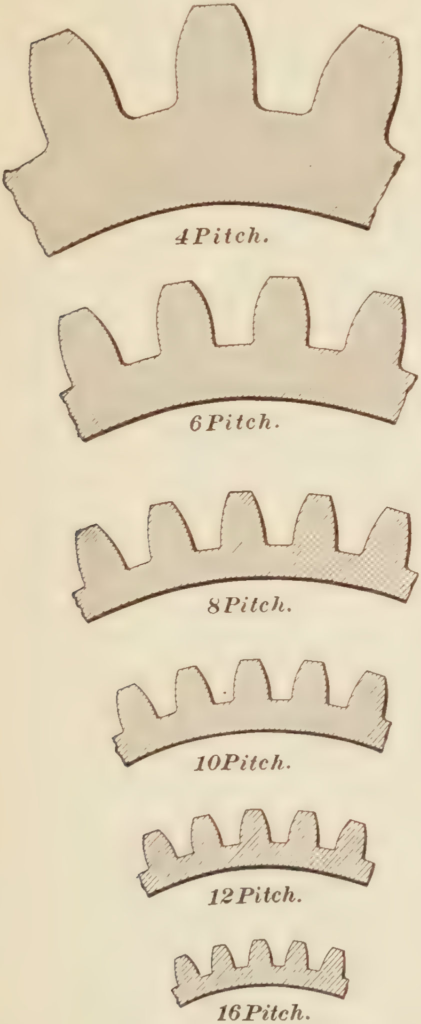


FIG. 415.

It would be very convenient to have the circular pitch expressed in manageable numbers like 1 inch, $\frac{3}{4}$ inch, etc.; but as the circumference of a gear is 3.1416 times its diameter, this requires awkward numbers for the diameters. Thus, a wheel of 40 teeth, 1 inch pitch, would have a circumference on pitch circle of 40 inches and a diameter of 12.732 inches. Of the two, it is more convenient in the great majority of cases to have the diameters expressed in numbers that can be easily handled. In order, however, to have the pitch in a convenient form also, a new pitch has been devised, *expressed in terms of the diameter* and called the **diametral pitch.**

1553. The diametral pitch is not a measurement like the circular pitch, but a *ratio*. *It is the ratio of the number of teeth in the gear to the number of inches in the diameter; or, it is the number of teeth on the circumference of the gear for one inch diameter of the pitch circle.* It is obtained by dividing the number of teeth by the diameter.

A gear, for example, has 60 teeth and is 10 inches in diameter. The diametral pitch is the ratio of 60 to 10 $= \frac{60}{10} = 6$, and the gear would be called a 6-pitch gear. From the definition, it follows that teeth of any particular diametral pitch are of the same size, and have the same width on the pitch line, whatever the diameter of the gear. Thus, if a 12-inch gear had 48 teeth, it would be 4 pitch. A 24-inch gear to have teeth of the same size would have twice 48 or 96 teeth, and $96 \div 24 = 4$, the same diametral pitch as before.

Fig. 415 shows the sizes of teeth of various diametral pitches.

Diametral pitch has also been defined as the number of teeth in a gear of one inch diameter, which amounts to the same as the definitions above.

Using for illustration a wheel 10 inches in diameter with 60 teeth, we have

$$\text{circular pitch} = \frac{\text{circumference}}{\text{No. of teeth}} = \frac{10 \times 3.1416}{60} = .524 \text{ inch.}$$

$$\text{Diametral pitch} = \frac{\text{No. of teeth}}{\text{diameter}} = \frac{60}{10} = 6.$$

1554. Other Definitions.—The other necessary definitions applying to the parts of a gear can be readily understood from Fig. 414. The thickness of the tooth and width of the space are measured on the pitch circle. A tooth is composed of two parts, the **addendum,** or outside of the pitch circle, and the **root,** which is inside.

A line through the outside end of the addendum is called the **addendum circle,** or **addendum line,** and one through the inside part of the root is called the **root circle,** or **root line.** The amount by which the width of the space is

greater than the thickness of the tooth is called the **back lash,** or **clearance.**

1555. Proportions for Gear Teeth.—With gears of large size, and often with cast gears of all sizes, the circular pitch system is used. In these cases, it is usual to have the addendum, whole depth, and thickness of the tooth conform to arbitrary rules based upon the circular pitch. None of these rules can be considered absolute, however. Machine-moulded gears require less clearance and back lash than hand-moulded, and very large gears should have less, proportionately, than smaller ones. The following table of proportions that have been used successfully will serve as an aid in deciding upon suitable dimensions. Column 1 is for ordinary cast gears, and column 2 is for very large gears having cut teeth. C stands for circular pitch.

TABLE 34.

	1	2
Addendum	$0.30C$	$0.30C$
Root	$0.40C$	$0.35C$
Whole Depth	$0.70C$	$0.65C$
Thickness of Tooth	$0.48C$	$0.495C$
Width of Space	$0.52C$	$0.505C$

The gears most often met with are the cut gears of small and medium size, like those, for example, on machine tools, which are almost invariably diametral pitch gears. The teeth are cut from the solid with standard milling cutters, proportioned with the diametral pitch as a basis. The system is also coming into very general use for cast gearing. In all diametral pitch gears, the addendum is made equal to 1 divided by the diametral pitch, and the working depth twice the addendum The end clearance is usually taken equal to $\frac{1}{8}$ of the addendum for cut gears, though the Brown & Sharpe Mfg. Co. use $\frac{1}{10}$ of the thickness of the tooth on the pitch line as the clearance. The side clearance, or back

lash, is made just enough to give a good working fit, and seldom exceeds $\frac{1}{50}$ of the pitch.

Using the above proportions, a 4-pitch gear would have the addendum $= 1 \div 4$, or $\frac{1}{4}$ of an inch; the working depth would be $2 \times \frac{1}{4} = \frac{1}{2}$ inch, and the clearance $\frac{1}{8} \times \frac{1}{4} = \frac{1}{32}$ inch. The whole length of the tooth would be $\frac{1}{2} + \frac{1}{32} = \frac{17}{32}$ inch. The thickness of the tooth would be $\frac{1}{2}$ the circular pitch, nearly. In a 10-pitch wheel the addendum would be $\frac{1}{10}$ inch and the length of the tooth $\frac{17}{80}$ inch; in a $2\frac{1}{2}$ pitch it would be $1 \div 2\frac{1}{2} = \frac{2}{5}$ inch and the length $\frac{17}{20}$ inch.

1556. Sizing Gear Blanks.—It is quite as important to be able to solve problems involving the diameter, number of teeth, and circular and diametral pitch of gear-wheels, as to be able to lay out the correct tooth curves. Several rules and examples will be given covering cases likely to be met with.

For convenience, these symbols will be used:

P = diametral pitch;
D = diameter of pitch circle;
$O\ D$ = outside diameter;
C = circular pitch;
N = number of teeth;
A = distance between centers of two wheels;
V = velocity—i. e., revolutions per minute.

When two wheels run together, small letters, like the above, will be used for the smaller wheel, the large letters applying to the larger wheel.

The product of the circular pitch of a gear and the diametral pitch is always the constant number 3.1416. *Hence, to change circular to diametral pitch, divide 3.1416 by the circular pitch; to change diametral to circular pitch, divide 3.1416 by the diametral pitch.* That is,

$$P = \frac{3.1416}{C}, \text{ and} \qquad \textbf{(149.)}$$

$$C = \frac{3.1416}{P}. \qquad \textbf{(150.)}$$

EXAMPLE.—If the circular pitch is 2 inches, the diametral pitch, or $P, = \frac{3.1416}{2} = 1.571$ inches, nearly. If the diametral pitch is 4, the circular pitch, or $C, = \frac{3.1416}{4} = .7854$ inch.

1557. Table 35 gives in the first two columns values of circular pitch corresponding to common values of diametral pitch, and in the last two columns values of diametral pitch corresponding to circular pitch values.

TABLE 35.

Diametral Pitch.	Circular Pitch.	Circular Pitch.	Diametral Pitch.
2	1.571 inches.	2 inches.	1.571
$2\frac{1}{4}$	1.396 "	$1\frac{7}{8}$ "	1.676
$2\frac{1}{2}$	1.257 "	$1\frac{3}{4}$ "	1.795
$2\frac{3}{4}$	1.142 "	$1\frac{5}{8}$ "	1.933
3	1.047 "	$1\frac{1}{2}$ "	2.094
$3\frac{1}{2}$	.898 "	$1\frac{7}{16}$ "	2.185
4	.785 "	$1\frac{3}{8}$ "	2.285
5	.628 "	$1\frac{5}{16}$ "	2.394
6	.524 "	$1\frac{1}{4}$ "	2.513
7	.449 "	$1\frac{3}{16}$ "	2.646
8	.393 "	$1\frac{1}{8}$ "	2.793
9	.349 "	$1\frac{1}{16}$ "	2.957
10	.314 "	1 "	3.142
11	.286 "	$\frac{15}{16}$ "	3.351
12	.262 "	$\frac{7}{8}$ "	3.590
14	.224 "	$\frac{13}{16}$ "	3.867
16	.196 "	$\frac{3}{4}$ "	4.189
18	.175 "	$\frac{11}{16}$ "	4.570
20	.157 "	$\frac{5}{8}$ "	5.027

The product of the diameter and diametral pitch in any gear is equal to the number of teeth, or

$$N = DP. \qquad \textbf{(151.)}$$

Also, knowing the number of teeth and the diametral pitch, the diameter may be found from the same formula.

EXAMPLE.—(*a*) If a wheel is 30 inches in diameter and 3 pitch, how many teeth has it?

(*b*) Of what diameter is a $2\frac{1}{2}$ pitch gear having 20 teeth?

SOLUTION.—(*a*) $N = DP = 3 \times 30 = 90$ teeth. Ans.

(*b*) $D = \frac{N}{P} = \frac{20}{2\frac{1}{2}} = \frac{40}{5} = 8$ inches. Ans.

The outside diameter of a gear equals the pitch diameter, plus twice the addendum, or $O\,D = D + 2 \times \frac{1}{P}$. But $D = \frac{N}{P}$. Hence, to obtain the outside diameter, knowing the diametral pitch and number of teeth, we have

$$O\,D = \frac{N}{P} + 2 \times \frac{1}{P} = \frac{N+2}{P}. \qquad \textbf{(152.)}$$

EXAMPLE.—A wheel is to have 48 teeth, 6 pitch; to what diameter must the blank be turned?

SOLUTION.—By formula **152**, $O\,D = \frac{N+2}{P} = \frac{48+2}{6} = 8.333$ inches. Ans.

EXAMPLE.—A gear blank measures $10\frac{1}{2}$ inches in diameter and is to be cut 4 pitch. How many teeth should the gear cutter be set to space?

SOLUTION.—From formula **152**, $O\,D = \frac{N+2}{P}$, or $N = O\,D \times P - 2 = 10\frac{1}{2} \times 4 - 2 = 42 - 2 = 40$ teeth. Ans.

To find the diameter, the circular pitch and number of teeth being given, we have, from the definition of circular pitch,

$$D = \frac{C \times N}{3.1416}. \qquad \textbf{(153.)}$$

EXAMPLE.—What is the diameter of a gear-wheel which has 75 teeth and whose circular pitch is 1.675 inches?

SOLUTION.—$D = \frac{1.675 \times 75}{3.1416} = 40$ inches. Ans.

Having given the distance between the centers of two gears and their velocities, the formulas for their diameters may be derived as follows:

From formula **136,** $VD = vd$, or $D = \frac{vd}{V}$, but $A = \frac{D + d}{2}$, or $D = 2A - d$. Equating values of D,

$$\frac{vd}{V} = 2A - d,$$

$$\text{whence } vd = 2AV - dV,$$

$$\text{or } d(v + V) = 2AV,$$

$$\text{and } d = \frac{2AV}{V + v}, \qquad \textbf{(154.)}$$

where V = velocity of the large gear and v that of the small gear.

In like manner,

$$D = \frac{2Av}{v + V}. \qquad \textbf{(155.)}$$

EXAMPLE.—Given the distance between centers of two gears = $5\frac{1}{2}$ inches. What must be their diameters so that the ratio of their speeds will be as 3 is to 1?

SOLUTION.—By formula **154,** $d = \frac{2 \times 5\frac{1}{2} \times 1}{1 + 3} = 2\frac{3}{4}$ inches.

By formula **155,** $D = \frac{2 \times 5\frac{1}{2} \times 3}{3 + 1} = 8\frac{1}{4}$ inches. Ans.

EXAMPLES FOR PRACTICE.

1. (*a*) How many teeth has a $2\frac{1}{2}$-pitch gear, 4 feet in diameter? (*b*) What is the circular pitch of this gear?

Ans. { (*a*) 120 teeth.
(*b*) 1.257 inches.

2. What is the outside diameter of a gear blank from which a wheel is to be cut having 50 teeth 4-pitch? Ans. 13 inches.

3. The pitch diameter of a gear is 25 inches. What is its outside diameter, supposing it to be 6-pitch? Ans. 25.333 inches.

4. A gear blank measures 10.2 inches in diameter and is to be cut 10-pitch. How many teeth should the gear cutter be set to space?

Ans. 100 teeth.

5. Given the distance between the centers of two gears = 20″. What must be their diameters so that the ratio of their speeds will be as 6 : 5? Ans. 18.181 inches and 21.818 inches.

1558. Law of Tooth Contact.—The **pitch point** of two gears is the point of contact C, in Fig. 416, of the pitch lines. It is the point at which the line of centers $O\,O'$

intersects the pitch circles. The **point of contact** is the point where two teeth touch each other. In order that two gear-wheels may have the same relative velocities at every point as their corresponding pitch cylinders, the tooth curves must be of such a shape that at the point of contact they will both be at right angles to a line $N N$, Fig. 416, passing through the pitch point and point of contact. This line is called the **common normal** to the tooth curves.

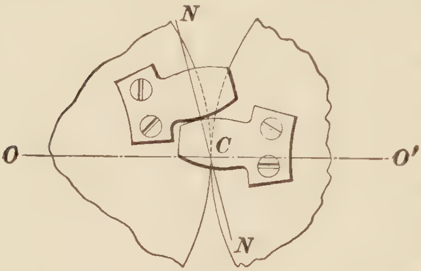


FIG. 416.

1559. The **path of contact** is the curve described by the point of contact during the entire action of a pair of teeth. This curve always passes through the pitch point.

1560. Angle and Arc of Action.—The angle through which a wheel turns from the time when one of its teeth comes in contact with a tooth of the other wheel until the point of contact has reached the line of centers is the **angle of approach**; the angle through which it turns from the instant the point of contact leaves the line of centers until the teeth are no longer in contact is the **angle of recess.** The sum of these two angles forms the **angle of action.** The arcs of the pitch circles which measure these angles are called the arcs of **approach, recess,** and **action,** respectively.

In order that one pair of teeth shall be in contact until the next pair begin to act, the arc of action must be at least equal to the pitch.

THE EPICYCLOIDAL SYSTEM.

1561. The Tooth Outline.—In Fig. 417, let O and O' be the centers of two pitch circles, in contact at the pitch point C; and let a smaller circle, whose center is at o, be tangent to both circles at C. Suppose the three centers to

be fixed, and the circles to move in rolling contact with each other in the direction of the arrows, the circle o carrying a marking point E. E will then describe a curve $E\ d$ on the plane of the circle O, and a curve $E\ e$ on the plane of the circle O'. $N\ N$, the common normal of these curves, will pass through the pitch point C, so they are suitable for tooth outlines.

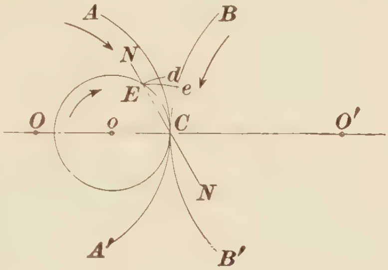


FIG. 417.

NOTE.—In mathematics, the words normal and perpendicular have the same meaning.

It will be observed that the relative motions of the circles are the same as though the small one rolled on the outside of O' and on the inside of O. $E\ e$ is, therefore, the epicycloid of $B\ B'$, and would answer for the face of a tooth of O', while $E\ d$ is the hypocycloid of $A\ A'$, and would serve as a flank for a tooth of O. In like manner, faces for O and flanks for O' could be generated by a circle inside of $B\ B'$.

Since the method of rolling up and using these curves was fully described in the subject of Mechanical Drawing, more will not be said here concerning the process.

1562. Interchangeable Wheels.—It is not necessary that the two generating circles used should be of the same diameter; provided that the flanks and faces which act on each other are generated by the same circle as in the previous case. It is customary, however, to use the same size circle for faces and flanks of both wheels, and where it is desired to make a set of gears, any two of which will run together, the same size circle must be used for all.

1563. In Figs. 418 to 420 are shown the effects of different sizes, describing circles upon the flanks of the teeth. In the first, the circle is half the pitch circle, and the flanks described are radial. In the second, with a smaller circle, the flanks curve away from the radius, giving a strong tooth, and in the third, with a larger circle, the flanks curve

inwards, giving a weak tooth, and one difficult to cut. It would seem, therefore, that a suitable diameter for the describing circle would be one-half the pitch diameter of

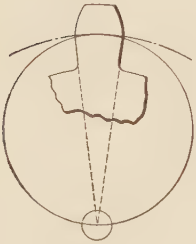


FIG. 418.

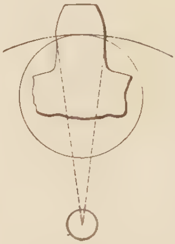


FIG. 419.

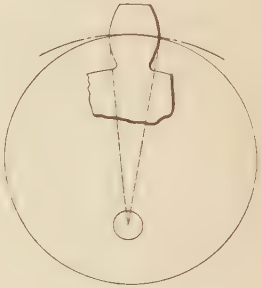


FIG. 420.

the smallest wheel of the set, or one-half the diameter of a 12-tooth pinion, which, by common consent, is taken as the smallest wheel of any set.

It has been found, however, that a circle of five-eighths the diameter of the pitch circle will give flanks nearly parallel, so that teeth described with this circle can be cut with a milling cutter. For this reason, some gear cutters are made to cut teeth based upon a describing circle of five-eighths the diameter of a 12-tooth pinion, or one-half the diameter of a 15-tooth pinion.

It is more common practice to take the describing circle equal to one-half the diameter of a 12-tooth than of a 15-tooth pinion, and this is the size used in this Course.

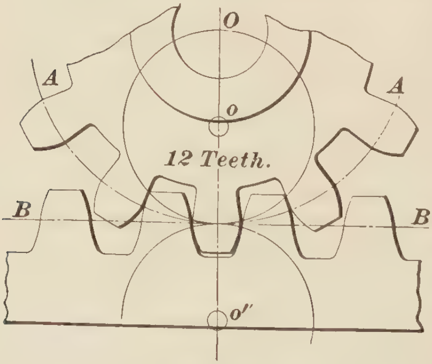


FIG. 421.

1564. Rack and Wheels.—A rack may be considered as a wheel having an infinite diameter. The pitch line of a rack is, therefore, a straight line, and for every revolution of the wheel the rack will travel a distance

equal to the circumference of the wheel. The construction of the tooth is shown in Fig. 421, describing circles for the rack teeth rolling on the line *B B*, and forming cycloids. If both circles are of the same diameter, and are the same as used for generating the tooth curves for an interchangeable series of wheels, the rack will evidently mesh with any of the wheels.

1565. Annular, or internal, gears are those having teeth cut on the inside of the rim. The width of space of an internal gear is the same as the tooth of a spur gear. Two describing circles are used as before, and, if they are of equal diameter, the gear will interchange with spur wheels for which the same describing circle was used.

In Fig. 422 is represented an internal gear with pitch circle *A A*, inside of which is the pinion with pitch circle *B B*. The generating circle *O*, rolling inside of *B B*, will

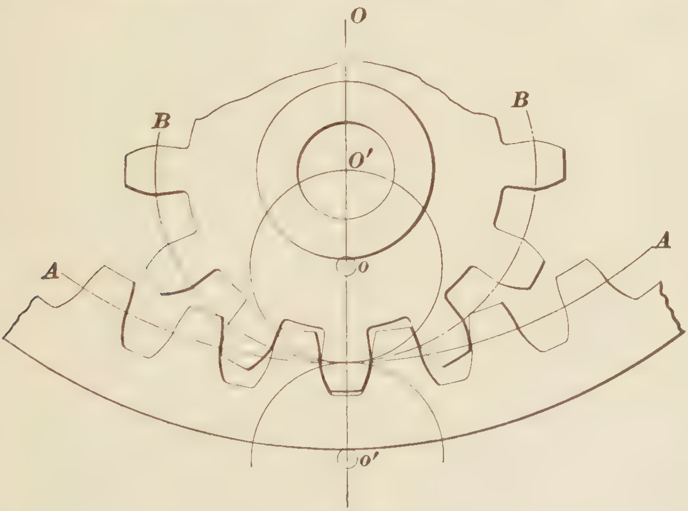


FIG. 422.

describe the flanks of the pinion, and rolling inside of *A A*, faces for the annular wheel. Similarly, the corresponding faces and flanks will be described by *o'*. The only special rule to be observed in regard to epicycloidal internal gears

is that *the difference between the diameters of the pitch circles must be at least as great as the sum of the diameters of the describing circles.*

This is illustrated by Fig. 423. A is the pitch circle of an internal gear, and B of the pinion. Then, for correct action, the difference ($D-d$) of the diameters must be at least as great as c, the sum of the diameters of the describing circles. To take a limiting case, suppose A to have 36 teeth and B 24 teeth. A wheel with a diameter equal to $D-d$, as shown dotted at E, would, therefore, have 36, minus 24, teeth, or 12 teeth. In the 12-tooth interchangeable system, this latter would be the smaller wheel of the series, and the describing circles would be half its diameter. From this it follows that, if $D-d$, the diameter of E, is not to be exceeded by the sum c of the diameters of the describing circles, B is the largest wheel that can be used with A. *Hence, when the interchangeable system is used, the number of teeth in the two wheels must differ by at least the number in the smallest wheel of the set.* If we wished, for example, to have 18 and 24 teeth, and the gears were to interchange, describing circles, half the diameter of a 6-tooth pinion, would be used, this being taken as the smallest wheel.

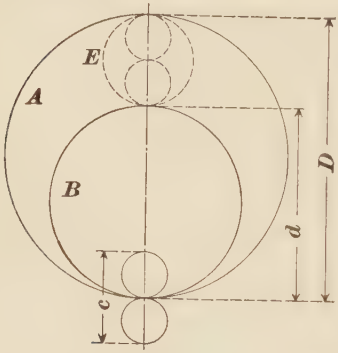


FIG. 423.

THE INVOLUTE SYSTEM.

1566. Let O and O' be the centers of two cylinders that are a short distance apart, and $D E$ a cord that has been wrapped several times around them in opposite directions, as shown in Fig. 424. If the circle $D D'$ be turned in the direction of the full arrow, the cord $D E$ will cause the cylinder $E E'$ to turn also in the opposite direction, as shown by the full arrow, and points on the cord $D E$ will describe

portions of the involute curve. In order to better comprehend this, imagine a piece of paper to be attached to the bottom of each cylinder, as shown, and that the width of each piece is the same as the distances between the cylinders. Now, suppose that a pencil be attached to the cord at E, the point of tangency of the line $D\ E$ with the cylinder $E\ E'$, in such a manner that it can trace a line on the piece of paper attached to the cylinder $E\ E'$, if a proper motion be given to the cord $D\ E$. Turn the cylinder $D\ D'$ in the direction of the arrow, i. e., rotate it left-handed. The point E will travel towards the cylinder $D\ D'$ in the straight line $E\ D$, and gradually diverge from the cylinder $E\ E'$. During this movement, the pencil attached at E will trace the involute curve m, shown dotted on the piece of paper. In the same manner, if the pencil be attached at D, and the cylinder $E\ E'$ be rotated in the direction of the dotted arrow, the dotted involute m' will be traced on the piece of paper attached to the cylinder $D\ D'$. Suppose that part of the pieces of paper to the right of the curve m and to the left of the curve m' be removed and that $E\ E'$ be rotated until the curve m takes the position $e\ d$; also, that $D\ D'$ be rotated until m' takes the position $g\ h$, the two curves being tangent to each other at k on the line $D\ E$. The cord $D\ E$ will be at right angles to both curves m and m' at the point of contact. The curves will, therefore, be suitable tooth curves, according to the law of tooth contact, if $E\ D$

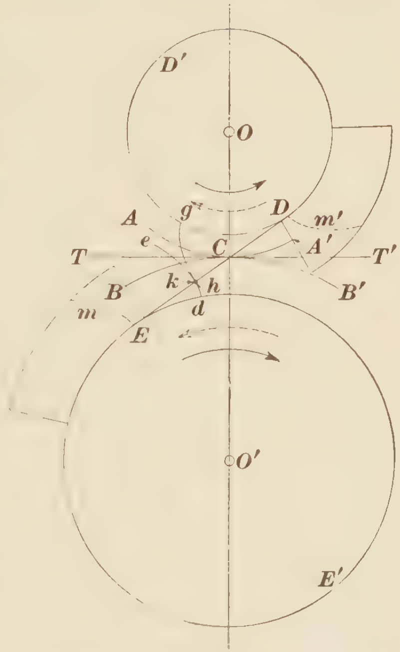


FIG. 424.

always passes through the pitch point. To make it do this, it is simply necessary to connect the two centers by the line $O\ O'$, and through its point of intersection C with $E\ D$ to draw the pitch circles $A\ A'$ and $B\ B'$. Two gears, therefore, with pitch circles $A\ A'$ and $B\ B'$, and with involute teeth formed from circles the size of $D\ D'$ and $E\ E'$, will have the same relative velocity as two pitch cylinders with radii $O\ C$ and $O'\ C$ in rolling contact. Such gears are sometimes called **single-curve gears,** because a single involute curve serves for both face and flank.

If the centers O and O' should now be moved apart so that the pitch circles do not touch, the relative velocities of $D\ D'$ and $E\ E'$ would evidently remain unchanged, since they are connected by the cord $E\ D$. The curves described by point k would also be the same, because they would still be involutes of the same circles. From this, it follows that the distance between the centers of involute gears may be varied without disturbing their relative velocity or the action of the teeth—a property peculiar to the involute system.

1567. Standard Gears.—In Fig. 424, let a line $T\ T'$ be drawn at right angles to $O\ O'$. Then, the angle $D\ C\ T'$, made by $E\ D$ with $T\ T'$, is called the **angle of obliquity,** and the circles $D\ D'$ and $E\ E'$, from which the curves are derived, and which are tangent to $E\ D$, are called the **base circles.** In standard interchangeable gears, based upon the diametral pitch, the angle of obliquity is taken at 15°, which brings the distance between the base circle and the pitch circle at about $\frac{1}{60}$ the pitch diameter. It would be well to use these values for all gears.

1568. Fig. 425 shows two standard gears, $A\ A$ and $B\ B$ being the pitch circles, and $D\ D'$ and $E\ E'$ the base circles. $T\ T$ was drawn through C at right angles with $O\ O'$, and $N\ N$ was drawn through C, making an angle of 15° with $T\ T$. The path of contact is along the straight line $N\ N$, and the distance along $N\ N$ from a point on one tooth to a corresponding point on the next is called the

normal pitch. The parts of the teeth above the base circle are involutes, and the flanks below the base circle are radial.

1569. Interference takes place when a part of one tooth crowds another at some point during the action, so that the gears will not run. To determine whether any pair of involute gears will work well together, draw lines $O\ D$ and $O'\ E$, Fig. 425, perpendicular to the line of action $N\ N$. So long as the intersections of the addendum circles (shown dotted) and the line of action fall between points E and D, as at e, there will be no interference. If they fall outside, as at d, both wheels will interfere, while, if the

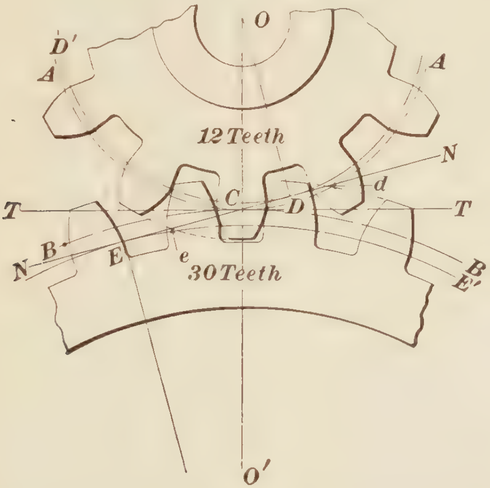


FIG. 425.

addendum circle of only one gear cuts the line of action outside, the teeth of that gear will interfere. Interference can be avoided by slightly rounding the ends of the teeth on the larger wheels, the amount to be determined by drawing the teeth in different positions. In the interchangeable system, all the gears are made to run with the smallest one of the set by giving epicyclodial points to the interfering teeth, so that they will work smoothly with the radial flanks.

1570. The **smallest wheel** in the interchangeable series has 12 teeth, the same number as in the epicycloidal

system. The reason for this is that it is the smallest wheel having a contact of the parts of the teeth which are true involute curves during an arc of action, equal to the circular pitch. Wheels of ten teeth will run together, however, although the action is not correct.

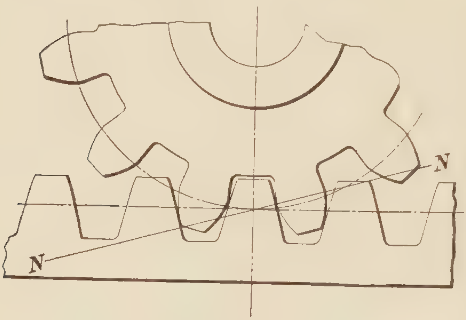


FIG. 426.

1571. The **involute rack** (see Fig. 426) has the sides of the teeth at an angle of 15° with the pitch line, or perpendicular to the line of action $N\,N$. The ends of the teeth should be rounded to run with the 12-tooth pinion.

1572. Internal Gears.—The construction is shown in Fig. 427. The obliquity (= 15°) is $T\,C\,N$, and the base

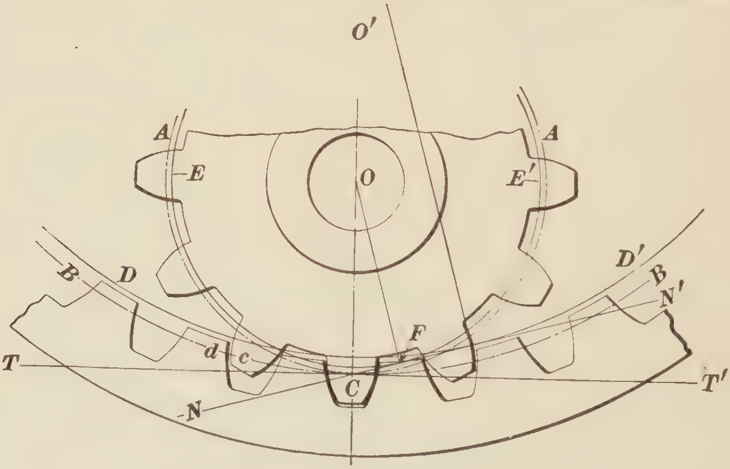


FIG. 427.

circles $E\,E'$ and $D\,D'$ are drawn tangent to the line of action $N\,N'$, about the centers O and O', respectively. The addendum circle for the internal gear should be drawn

through F, the intersection of the path of contact $N N'$ with the perpendicular $O F$ drawn from the center of the pinion. The wheel will then be nearly, or quite, without faces, and the teeth of the pinion, to correspond, may be without flanks. If the two wheels are nearly of the same size, points c and d will interfere, which can be avoided by rounding the corners.

1573. In General.—Formerly, the epicycloidal system was used almost exclusively, but of late the involute is rapidly gaining in favor. Its distinctive features are, the adjustability of the centers of the wheel and the great strength of the tooth. The chief objection that has been raised against involute teeth is the obliquity of action, causing increased pressure upon the bearings. Where the obliquity does not exceed 15°, however, this objection is not a serious one.

BEVEL GEARS.

1574. In drawings of spur gears, the tooth curves in the epicycloidal system are obtained by causing the generating circles to roll upon the pitch circles. The tooth curves, however, represent curved surfaces perpendicular to the plane of the paper and the pitch circles, and generating circles represent the ends of cylindrical surfaces which are in rolling contact. It may be assumed, therefore, that tooth surfaces are generated directly by generating *cylinders* rolling upon pitch *cylinders*. In bevel gearing, the pitch surfaces are cones, which, when in rolling contact, have their apexes at a common point, and it may be assumed that the tooth surfaces are generated by generating *cones*, rolling upon the pitch cones.

In spur gearing, the teeth of two wheels bear along straight lines, which are perpendicular to the plane of the paper. In bevel gearing, the teeth are in contact along straight lines, *but these lines are perpendicular* to the surface of a sphere, and all of them pass through the center of the sphere, which is at the point where the apexes of the two pitch cones meet. That this is the case will now be explained.

1575. In Fig. 428, let $C\ O\ B$ represent a pitch cone, the part $C\ D\ E\ B$ being the pitch surface of a bevel gear, and let $A\ O\ C$ be the generating cone. If we suppose the generating cone to describe the tooth surface $m\ n\ o\ p$ by rolling upon the pitch cone, the line $n\ o$, representing the outer edge of the tooth, will lie upon the surface of a sphere whose radius is $O\ n$. For the point n, which describes this line, is always at a fixed distance from the center O; hence, every point in the line $n\ o$ is equally distant from O, and as, in a spherical surface, every point is equally distant from a point within called the center, it follows that $n\ o$ must lie upon a spherical surface.

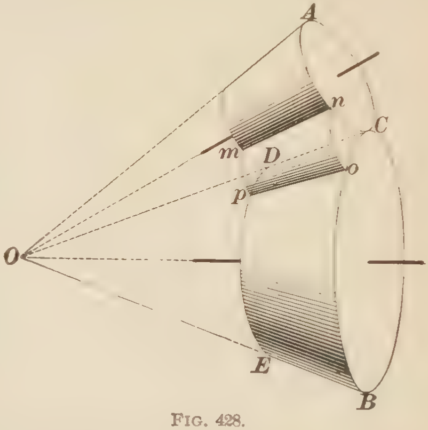


FIG. 428.

1576. To be theoretically exact, therefore, the tooth curves for bevel gears should be traced upon the surface of a sphere, as shown in Fig. 429. This method is not a practical one, however, and would have no advantage over what is known as **Tredgold's approximation,** which is much simpler and is universally used.

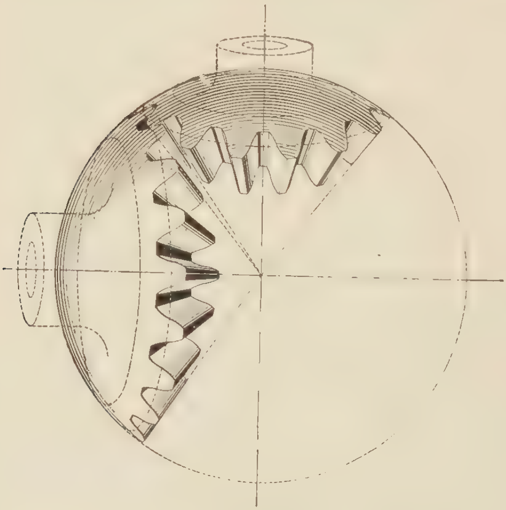


FIG. 429.

By this method, the tooth curves are drawn on cones tangent to the spheres at the pitch lines of the gears, as shown in Fig. 430. The process is simply to develop or unwrap the surfaces of the cones, the unwrapped surfaces being represented by $A\ B\ C$ and $C\ D\ E$ in the figure. The length of the arc $A\ B\ C$ is equal to the length of the pitch circle $A'\ C$, and the arc $C\ D\ E$ is equal to the pitch circle $C\ E'$. The gear teeth are then drawn upon the unwrapped surfaces, precisely

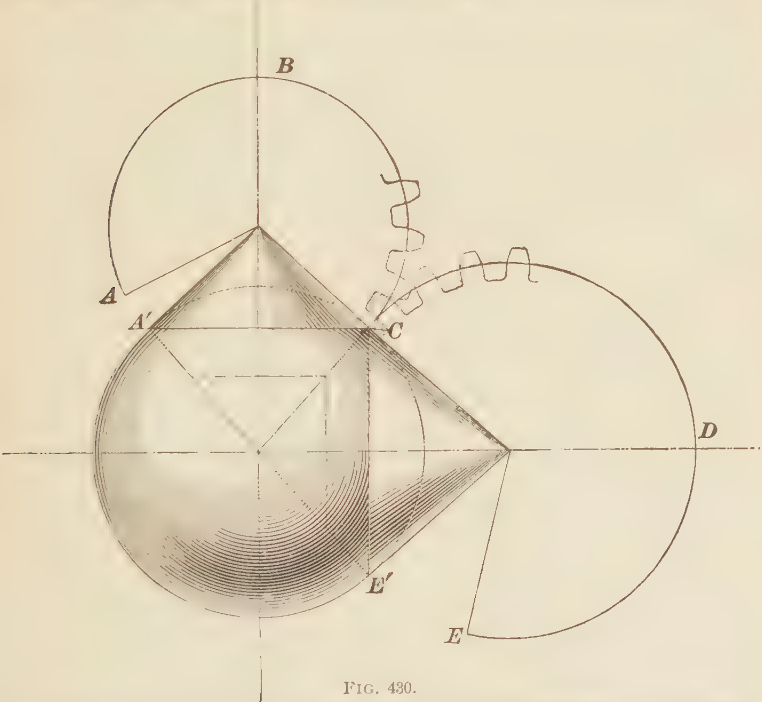


FIG. 430.

as for spur gears of the same pitch and diameter. This process is fully described in the subject of Mechanical Drawing.

The teeth, as laid out by Tredgold's method, will vary somewhat from the shape of the spherical teeth, though usually the variation is slight. The actual error of the system, however, is less than this difference, for though the tooth curves on each gear may not be the same as those on the sphere, the amount of their divergence from perfect curves to transmit a uniform motion will be of no practical importance.

WORM GEARING.

1577. A **worm** is a screw made to mesh with a wheel called a **worm-wheel**, the two forming **worm gearing**, or **screw gearing.** Worm gearing possesses the following characteristics:

I.—The velocity ratio depends upon the pitch of the screw, i. e., the distance the screw advances in one revolution, and not upon the diameters of the pitch cylinders. *If the worm is single-threaded, it must make as many turns as there are teeth in the wheel for every revolution of the latter; if double-threaded, it will make one-half as many turns.*

II.—The direction in which the worm-wheel turns depends upon whether the worm has a right-hand or left-hand thread.

III.—The end thrust of the screw causes the motion of the wheel.

1578. Form of Teeth.—Fig. 431 illustrates a worm and wheel. It will be noticed that in the longitudinal section, taken through the worm, the threads appear to be like involute rack teeth. The worm is usually made in a screw-cutting lathe, and as it is easier to turn the threads with straight sides, it is better that they should be of the involute form. Involute teeth should then be used on the wheel of a pitch to correspond with the threads on the worm.

1579. Pitch.—The circular-pitch system is almost universally used for worm gearing, because lathes are seldom provided with the correct change gears for cutting diametral pitches. It is not so inconvenient, however, in the case of worm gearing as with the spur gearing. If the diameter of the worm-wheel should come in awkward figures, the diameter of the worm can be made such that the distance between centers will be any desired dimension. The circular pitch of the gear must equal the pitch of the worm.

1580. Close-Fitting Worm and Wheel.—To make a close-fitting wheel, a worm is made of tool steel and then fluted and hardened similar to a tap. It is almost a duplicate

of the worm to be used, being of a slightly larger diameter to allow for clearance. This cutter, or **hob,** is placed in mesh with the worm-wheel, on the face of which notches have been cut deep enough to receive the points of the teeth

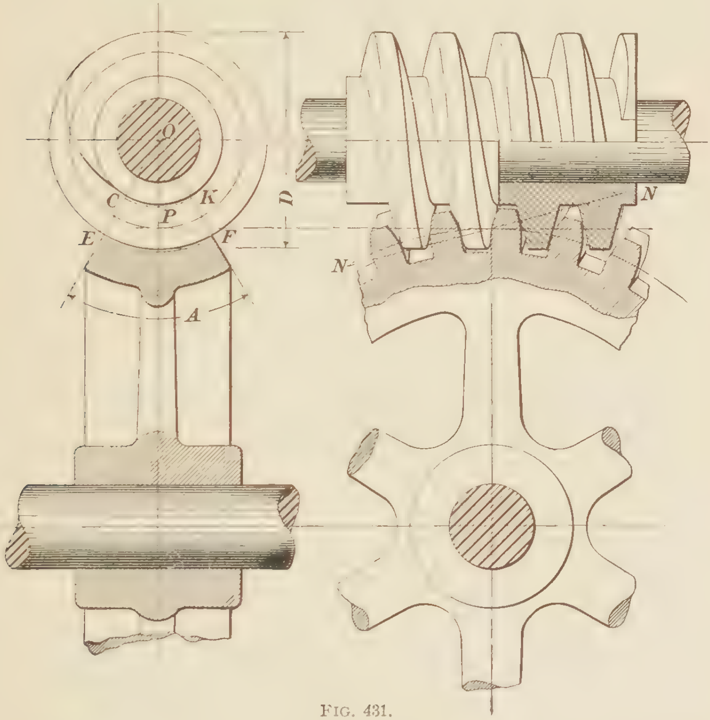


FIG. 431.

of the hob. The hob is then made to drive the wheel, and is dropped deeper into it at each revolution of the latter until the teeth are finished.

Fig. 431 represents a close-fitting worm and wheel. The pitch circles of each are in contact at P. The outside diameter D of the worm may be made four or five times the circular pitch. The arcs $C\,K$ and $E\,F$ are drawn about O and limit the addendum and root of the wheel teeth, the distance between them being the whole depth of the teeth. End clearance is allowed the same as for spur gearing. The

angle A is generally taken at either 60° or 90°. The whole diameter of the wheel blank can be obtained by measuring the drawing.

The object of hobbing a wheel is to get more bearing surface of the teeth upon the worm thread, making the outline of the teeth something like the thread of a nut.

1581. Worm-Wheels Like Spur Gears.—When worm-wheels are not to be hobbed, there is little to be gained by making the face of the wheel concave to fit the worm. It is better to construct the blanks like a spur-wheel blank. The teeth can then be cut in a straight path diagonally across the face of the blank to fit the angle of the worm thread.

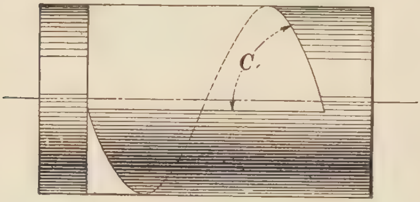


FIG. 432.

This angle may be obtained as follows: Fig. 432 represents a right-angled triangle cut out of paper and wrapped around a cylinder. The hypotenuse of the triangle forms the thread and, as the base is parallel with the axis of the cylinder, the angle of the worm thread is the angle C between the hypotenuse and the base of the triangle. *Hence, the tangent of the angle of the thread = the circumference of the cylinder, divided by the pitch.*

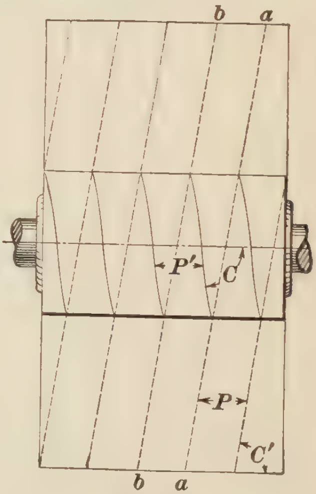


FIG. 433.

In Fig. 343 is shown a graphical method. The lines $a\ a$, $b\ b$, etc., are the development of the screw. Angle C' = angle C is the proper angle for the teeth of the worm-wheel. The distance $P' = P$, parallel to the axis of the screw, is the pitch.

1582. Interference.—The same rules apply here that were given under the involute system. When the worm is to work with a wheel having a smaller number of teeth than it was designed for, interference will occur. It can be avoided by rounding the tops of the threads of the worm, but it is easier to make the wheel blank somewhat larger than is called for by the number of teeth. The teeth will then have very short flanks, the action being almost entirely upon the faces, where interference cannot occur.

RATCHET WHEELS.

1583. Fig. 434 represents a ratchet wheel *A* turning upon the pin *O*. *C* is a vibrating lever carrying the **pawl, click,** or **catch** *B*, which acts upon the teeth of the wheel. As the arm moves back, or right-handed, the click lifts and slides over the points of the teeth; when it returns, the click drops against a tooth and carries the wheel with it.

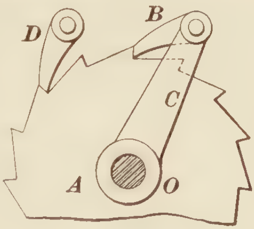


FIG. 434.

In case it should be desired to prevent the wheel from moving backwards when the click is moving backwards, a fixed pawl, similar to *D*, would be made to bear on the wheel and drop behind each tooth as it passed under. Here, *D* would allow a left-hand, but prevent a right-hand rotation of *A*.

1584. In Fig. 434, in order that the arm may produce motion in the wheel, its vibration must be at least sufficient to cause the latter to advance one tooth; but by arranging several clicks in the same lever it becomes possible to give a motion to the wheel corresponding to less than one tooth for each vibration of the arm. Fig. 435 shows such a construction, *B*, *B'*, and *B''* being proportioned so that they come into action alternately. Thus, when the wheel *A* has moved back an amount corresponding to one-third of a tooth, the click *B'* will be in contact with tooth

b', and, if the arm should then move the wheel forwards a distance of at least one-third a tooth and then return to its former position, click B'' would fall behind b'', ready to turn the wheel. Thus arranged, a slight motion is obtained and comparatively large teeth may be used. A neater construction than the above would be to put all the clicks on one pin side by side, in which case a wide wheel would be necessary.

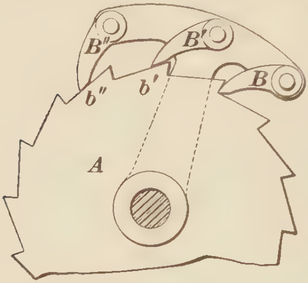


FIG. 435.

1585. Reversible Clicks.—In feed mechanisms, such as are used on shapers and planers operating on metal, and which must be driven in either direction, an arrangement like that in Fig. 436 is used. Wheel A has radial teeth, and the click, which is made symmetrical, can occupy either position B or B'. In order that the click may be held firmly against the ratchet wheel, its axis is provided with a small triangular piece, shown dotted, against which is a flat end-presser, always urged upwards by a spring (also shown dotted). Whichever position B may be in, it will be held against the wheel.

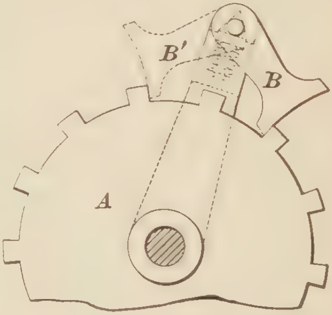


FIG. 436.

1586. Adjusting the Motion.—Ratchet wheels are largely used on machines requiring a "feed." In all such cases they must be so arranged that the feed can be easily adjusted. This is often done by changing the swing of the lever C, Fig. 434, which is usually connected by a rod with a vibrating lever, having a definite angular movement at the proper time for the feed to occur. This lever is generally provided with a **T** slot, in which the pivot for the rod can be adjusted by means of a screw and nut. By varying the

distance of the pivot from the center of motion, either one way or the other, the swing of the arm C can be regulated and the feed made to occur in either direction desired.

1587. Another method of adjusting the motion is shown in Fig. 437. The wheel turns upon a stationary shaft or stud O, and the end of the shaft is turned to a smaller diameter than the rest and is threaded, thus forming a shoulder against which an adjustable shield S can be clamped by the nut n. Back of the wheel is the arm, also loose on the shaft, carrying the click B, which latter should be of a thickness equal to that of the wheel, plus that of the shield. The teeth of the wheel may be made of a shape suitable to gear with another wheel to which the feed motion will then be imparted, or another wheel back of and attached to the one shown could be used for the purpose.

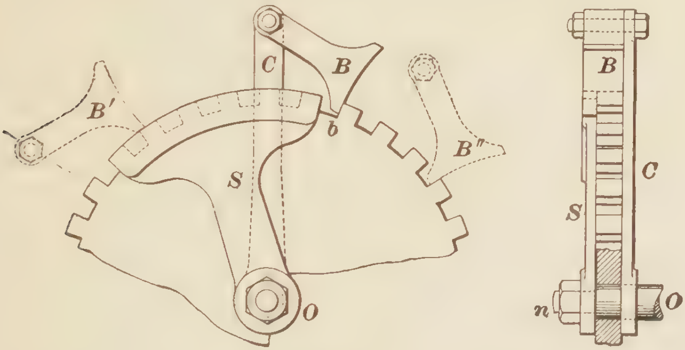


FIG. 437.

The extreme left-hand position of B is shown at B'. Here the click rides on the shield and does not come in contact with the teeth of the wheel. When the click comes to the right-hand edge of the shield, however, it will drop into contact at b; and, if B'' is the extreme right-hand position, the wheel will be turned through a space corresponding to three teeth. If the shield should be turned to the right, a smaller number of teeth would be moved each time; if it should be turned to the left, a feed of four or more teeth, up to the full capacity of the stroke, could be obtained; while, with

the shield in its mid-position, it would carry the click during the whole swing of the arm and there would be no feed.

1588. Ratchet and Screw.—Where ratchets are employed in the feed motions of machine tools, they are made to operate a screw, which in turn drives the "head" carrying the tool.

EXAMPLE.—A ratchet having 80 teeth is attached to the end of a screw having six threads per inch. If the click is set to move the ratchet three teeth for every stroke of the arm, how much "feed" would the tool have, supposing it to be moved directly by the screw?

SOLUTION.—One turn of the screw would move the tool $\frac{1}{6}$ inch. But for each stroke, the ratchet and, hence, the screw, moves $\frac{3}{80}$ of a turn, and the tool would travel $\frac{3}{80} \times \frac{1}{6} = \frac{1}{160} = .00625$ inch. Ans.

APPLIED MECHANICS.

(CONTINUED.)

DYNAMOMETERS.

1589. **Dynamometers** are instruments for measuring power. They are divided into two main classes—**absorption dynamometers** and **transmission dynamometers.**

1590. The most common form of **absorption dynamometers** is the **Prony brake,** which consists simply of a friction brake designed to absorb in friction and measure the work done by a motor or the power given out by a shaft.

1591. A **transmission dynamometer** is used to measure the power required to drive a machine or do other work; then, to determine the power required to run the shafting in a mill, a transmission dynamometer would be interposed between the shafting and the source of power, and by suitable belt connections the shafting would be driven *through* the dynamometer, from which the power could be determined.

PRONY BRAKE.

1592. Fig. 438 represents a simple and common form of Prony brake. It consists of two wooden blocks A and B that are clamped together upon a pulley P, by the bolts and thumb-nuts c, c. The same bolts clamp on arm L to the upper block, from which a scale pan, bearing a known weight W, is suspended. The distance R, from the center of the

pulley to the perpendicular through the point from which the scale pan is suspended is also known. The counterweight w should be so adjusted as to just balance the extra length of L on the right, and the weight of the scale pan.

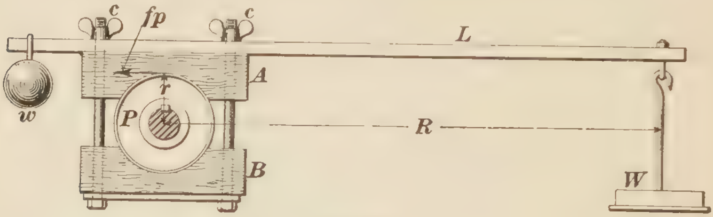


FIG. 438.

Suppose the pulley to revolve left-handed and the bolts c, c to be tightened until, with a weight W in the scale pan, the lever L will remain stationary in a horizontal position. *The foot-pounds of work absorbed by the brake can then be found by multiplying the weight W by the circumference of a circle whose radius is R (in feet) and by the number of revolutions of the pulley.*

It is important to note that neither the diameter of the pulley nor the pressure with which the blocks clamp the pulley enter into the calculations at all. For, letting p represent the pressure and f the coefficient of friction between the blocks and the pulley, the force at the face of the pulley tending to resist its rotation will be $f p$. The force tending to keep the lever L from turning, however, is W, and as the bolts are adjusted so that L remains constantly in a horizontal position, the moments of these two forces about the center of the pulley are equal, or $f p r = W R$.

1593. Now, the work done at the face of the pulley is equal to the force exerted × number of feet passed through, or, calling N the number of revolutions per minute, $2 \pi r \times f p \times N$. This, it will be seen, is the first member of the above equation multiplied by $2 \pi N$. Multiplying the second member by $2 \pi N$, also, to keep both members equal, we obtain as another expression for the work absorbed, $2 \pi R \times W \times N$. This is the formula used in calculations.

Hence, letting H. P. = number of horsepower absorbed;
R = length in feet of lever arm about center of shaft;
W = weight in scale pan;
N = number of revolutions per minute.

$$\text{H. P.} = \frac{2 \times 3.1416 \times R \times W \times N}{33,000}.$$

This may be reduced to

$$\text{H. P.} = .0001904\ R\,W\,N. \qquad \textbf{(156.)}$$

EXAMPLE.—A brake with an arm R, 6 ft. long, was placed on the fly-wheel of an engine. If the engine ran at 200 revolutions per minute, what power did it develop when the brake balanced with 14 pounds in the scale pan?

SOLUTION.—Applying formula **156**,
H. P. $= .0001904 \times 6 \times 14 \times 200 = 3.198$ horsepower. Ans.

1594. Brakes are often constructed of a metal band which extends entirely around the pulley, the rubbing surface being formed of blocks of wood fitted to the inside of the band. A weight arm is attached to one side of the pulley and the friction is varied by means of a bolt and nut used to connect the two ends of the band.

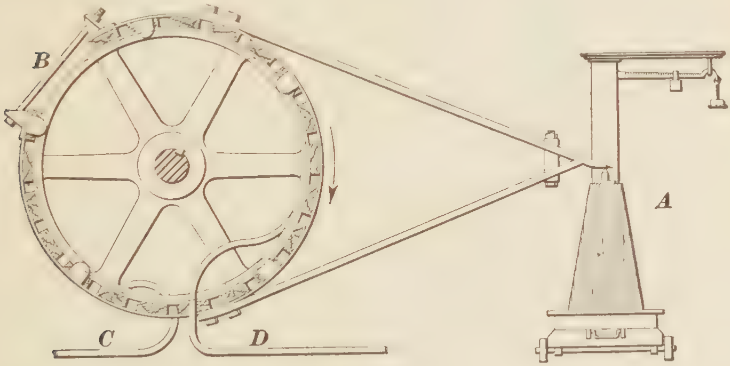


FIG. 439.

Instead of hanging weights in the scale pan the friction may be weighed on a platform scale, as shown in Fig. 439. In this case, the direction of rotation of both pulley and arm is the same.

It is essential that these brakes should be well lubricated, and for all except small powers means must be provided for conducting away the heat generated by friction. If there are internal flanges on the brake wheel, water can be run on to the inside of the rim, the flanges serving to retain the water at the sides and centrifugal force to keep it in contact with the rim. A funnel-shaped scoop can be used to remove

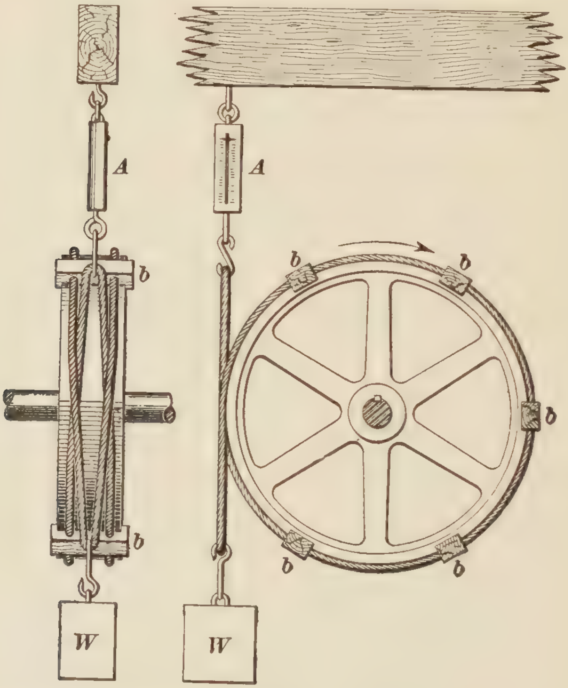


FIG. 440.

the water. It should be attached to a pipe and placed so as to scoop out the water, which should flow continuously. This arrangement is shown in Fig. 439.

1595. A rope brake, like that in Fig. 440, will give good results. The figure shows the construction so clearly that no description is necessary. To obtain the brake load, subtract the brake pull as registered by the spring balance from

the weight. In this case, the lever arm is r, equal to the radius of the pulley $+ \frac{1}{2}$ the diameter of the rope. If this radius be given in inches, formula **156** becomes

$$\text{H. P.} = .00001586\, r\, W N \qquad \textbf{(157.)}$$

TRANSMISSION DYNAMOMETERS.

1596. There are several forms of transmission dynamometers regularly manufactured, but, as rules for their use always accompany the machines, only one form will be described here.

The side and end elevations of the **differential transmission dynamometer** are shown in Fig. 441. W_1 is a

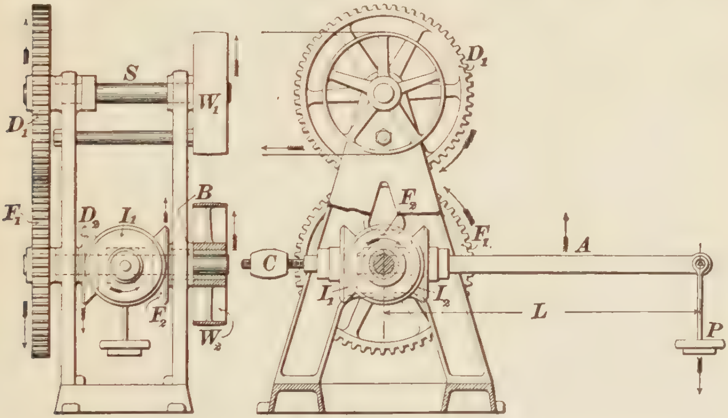


FIG. 441.

pulley to be belted up with the pulley on the line shaft, or other source of power, that has been driving the machine to be tested. W_2 (shown only in the end view), placed for convenience in the same plane with W_1, is of the same size as W_1, and is to be belted to the driving pulley on the machine. Connection between W_1 and W_2 is made by means of the shaft S, running in bearings on the frame, the two gears D_1, F_1, of equal diameters, and the differential gearing shown.

Of this latter, D_2 is keyed to the same shaft as F_1. F_2 is loose on the shaft and has a long hub reaching through the bearing B, and through the hub of the pulley W_2 to which

it is fastened. D_2 and F_2 are connected by the two miter gears I_1 and I_2 in the usual way, having their bearings on the arm A, and free to turn about the lower shaft. There is a scale pan at the right-hand end of the arm hung from a knife-edge, and it is clear that if a weight P be put in the pan sufficient to hold it down I_1 and I_2 will act simply as idlers, and F_2 will turn in the opposite direction and with the same speed as D_2. Hence, W_1 will turn in the same direction as W_2, and with the same speed.

A counterweight C is provided for balancing the arm when the dynamometer is at rest and there are no weights in the scale pan. To find the amount of power transmitted, the length L of the arm, the weight P in the scale pan, and the number of revolutions must be known. The arm is generally of such a length that the circumference which the knife-edge would describe if the arm revolved about the shaft would be some even number of feet, say ten.

From what has been stated before regarding bevel trains, it is evident that, if pulley W_1 is turned and F_2 is at the same time held fast so that it cannot turn, the arm will make one-half the number of revolutions made by W_1. Twice as great a weight will, therefore, be required in the scale pan to keep the arm stationary as would be necessary if the arm made the same number of revolutions. Hence, applying the principle of the Prony brake, and supposing that, in the circumference of the circle whose radius is L, the length of the arm is 10 feet, *two pounds in the scale pan will correspond to $1 \times 10 = 10$ foot-pounds of power transmitted per revolution of the shaft.*

EXAMPLES FOR PRACTICE.

(1) Given, lever arm of a Prony brake = 4.5 ft.; weight in scale pan, 2 lb. 4½ oz.; rev. per min. of pulley, 160. Required the power absorbed.

Ans. .313 H. P.

(2) A rope brake is used on a pulley 36" in diameter. The diameter of the rope is ⅝"; revolutions of pulley per minute, 200. How many horsepower are absorbed, the weight being 210 lb. and the balance reading being 5 lb.?

Ans. 11.9 H. P.

(3) An actual test with a rope brake showed a mean brake horsepower of 15.23. The mean number of revolutions of the wheel per

minute was 205, the weight used was 157 lb., and the mean back pull on the balance was 4 lb. What was the length of the lever arm?

Ans. 30.6 inches.

VALVE GEARS.

PLAIN SLIDE-VALVE.

1597. It is assumed that the student has studied carefully the discussion of valves and their relation to the action of the steam in the cylinder, as given in Art. **1232,** etc. We shall now treat more especially of the mechanics of valves and valve motions.

1598. Definitions.—For the purpose of review a few definitions will be repeated here in a brief form. They will also be convenient for reference.

The four principal points or events during one stroke of an engine are:

I.—The point of **admission** of steam to the cylinder.

II.—The point of **cut-off** of the steam from the cylinder.

III.—The point of **release** where the steam begins to be exhausted from the cylinder.

IV.—The point of **compression** where exhaust closes.

1599. Fig. 412 gives a sectional view of a plain slide or

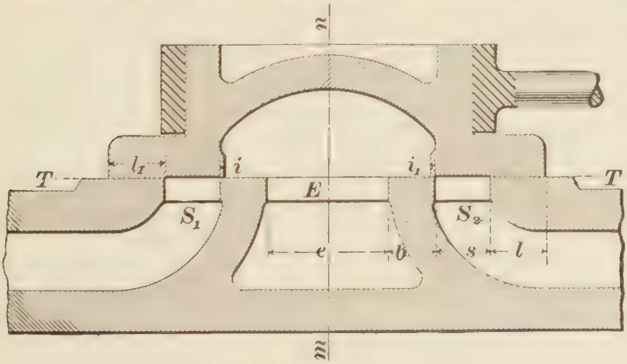


FIG. 412.

D valve. The under surface of the valve, called the **valve face,** slides over the **valve seat** T T on the cylinder. In

the cylinder are three **ports.** Two, S_1 and S_2, communicate with passages leading to the ends of the cylinder and are called **steam ports;** the third, E, leads to the atmosphere or condenser, and is termed the **exhaust port.**

In Fig. 442, the valve is in *mid-position* with its center n in line with the center m of the exhaust port E.

1600. The **lap** is the amount by which the edge of the valve overlaps the adjoining edge of the steam port when the valve is in mid-position. It is called the **outside** or **inside lap,** according as we refer to the outside or inside of the valve, l or i and l_1 or i_1 in the figure.

1601. **Lead** is the amount of the opening of the steam port at the beginning of the piston's stroke. The **lead angle** is the angle made by the center line of the crank with the center line of motion of the engine at the point of admission.

1602. The **displacement of the valve** is the distance that the center of the valve has moved from its mid-position.

1603. The **travel of the valve** is the total distance the valve moves one way. The **stroke** of an engine and **travel** of a valve are relatively similar terms.

1604. The valve is moved by an eccentric whose **throw** is equal to the diameter of the circle described by the center of the eccentric as it turns with the shaft. The radius of this circle is known as the **eccentricity.** The throw of the eccentric and travel of the valve are the same, if there is no intervening rocker which increases or decreases the action of the eccentric.

1605. The **angle of advance** of the eccentric is the angle by which the center line of the eccentric stands away from a line at right angles to the center line of the crank. The angle of advance is sometimes called *angular advance.* It is equal to the angle due to the lap + the angle due to the lead.

1606. Direction of Rotation.—The study of the slide-valve is essentially a study of the relative motions of the piston of the engine and the valve. The first thing to understand is the direction in which the crank will turn. This depends upon the way the eccentric is connected with the valve and upon the location of the angle of advance.

The eccentric may be connected with the valve in three ways: First, the eccentric rod may act directly upon the valve spindle; second, it may act through a rocker pivoted at one end, of the nature of a bell-crank; third, it may act through a reversing rocker pivoted near the center. In the first two instances, the valve will move with the eccentric, and the connection may be said to be direct. *In these cases, the eccentric will always be in advance of the crank, in the direction in which the crank is to turn, by an angle equal to 90° + the angle of advance.* That is, the crank will follow the eccentric.

The action of a reversing rocker is simply to cause the valve to move in a direction opposite to that in which the eccentric is moving. *Hence, when a reversing rocker is used, the eccentric will be behind the crank by an angle equal to 90° — the angle of advance.* That is, the crank will lead, and the eccentric will take positions exactly opposite to those in the previous case.

In what follows, the eccentric rod will be assumed to act directly on the valve spindle, unless stated to the contrary.

1607. Displacement of the Valve.—The next thing to consider is the position of the valve for any given position of the piston. Suppose at first that the crank and valve are actuated by slotted cross-heads like the one in Fig. 350, Art. **1442**, instead of by the usual connecting-rod and eccentric. Then, the displacement of the valve for any given piston position can be found as follows:

Draw the outer semicircle $A\ B\ C$ (Fig. 443) on one side of the stroke line $A\ C$, with a radius $O\ R$, equal to the length of the main crank. This will represent the path of the crank-pin R during one stroke. From the same center, draw the

inner semicircle with a radius *O r*, equal to the length of the valve crank. Now, suppose the piston to have moved along its stroke *A C* a distance of *A n*. The crank will then be in the position *O R*, *R* being perpendicularly above *n*. If we let *a* equal the angle of advance, the valve crank will

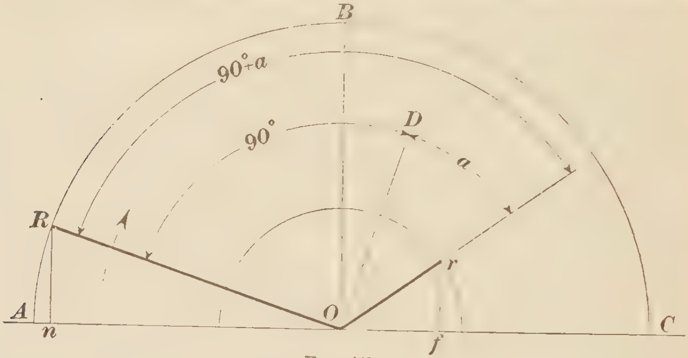


FIG. 443.

be at *O r*, ahead of the main crank by the angle *r o R*, equal to $90 + a$, and it is evident that the displacement will then be equal to *O f*. By valve crank is meant an imaginary crank, supposed to replace the eccentric, which has a radius equal to the radius of the eccentric.

1608. A more convenient method of finding the displacement is shown in Fig. 444. It forms the basis of a

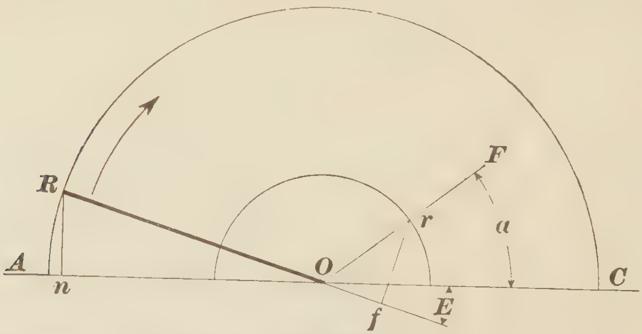


FIG. 444.

valve diagram, which will be explained later, and by which a slide-valve can be correctly proportioned, and examples involving lead, lap, angular advance, etc., can be solved.

Let the crank-pin paths be drawn as before on one side of the stroke line $A\ C$. From O, draw the line $O\ F$, making an angle a, with $O\ C$ equal to the angle of advance. Then, the length of a perpendicular drawn from r, the point of intersection of $O\ F$ and the inner circle, to the center line of the main crank, in whatever position it may be, will represent the displacement of the valve. In this case, the crank position is $O\ R$, and $r\ f$, perpendicular to the center line produced beyond O, is the displacement.

1609. Effect of Connecting-Rod.—Now, suppose that an engine has the crank and valve moved by a connecting-rod and eccentric, as in practice. It has previously been shown that in a crank motion, the longer the connecting-rod compared with the crank the more nearly the motion will approach that given by a slotted cross-head. Since in most engines the eccentric rod is very long compared with the eccentricity, the relative positions of the valve and cranks can be determined with sufficient accuracy by the foregoing method. The connecting-rod, however, is ordinarily only from four to six times the length of the crank. Hence, if it be required to accurately find the relative positions of the *crank* and *piston*, and hence the *valve* and *piston*, the effect of the connecting-rod should be taken into account.

1610. EXAMPLE.—Given, the length of the stroke of an engine, the travel of the valve, the angle of advance, and the length of the connecting-rod. What are the valve displacements at half stroke each way?

SOLUTION.—Describe the crank circle (Fig. 445) $A_1\ R_1\ C_1\ R_2$, with a radius equal to half the stroke. About the same center O describe the

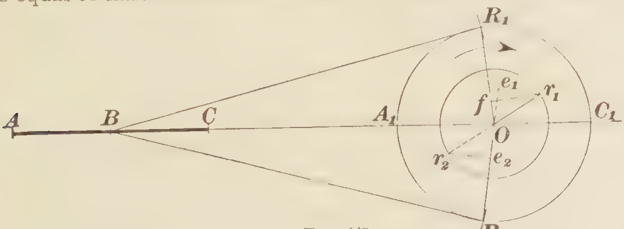


FIG. 445.

inner circle $r_1\ r_2$, which we will call the eccentric circle, with a radius equal to half the valve travel, or eccentricity. Draw the line of motion

$A\,C_1$, and on it lay off $A\,A_1$, $B\,O$, and $C\,C_1$, each equal to the length of the connecting-rod, giving the stroke $A\,C$ and the mid-position B of the piston. With B as a center and a radius equal to $B\,O$, cut the crank circle at R_1 and R_2, which will give the crank positions $O\,R_1$ and $O\,R_2$, corresponding to the mid-positions of the piston.

From O, draw $O\,r_1$, making an angle with $O\,C_1$ equal to the angle of advance of the eccentric, and intersecting the eccentric circle at r_1. Then, the perpendicular $r_1 f$, from r_1 to the crank position $O\,R_1$, is the displacement of the valve for that crank position. Likewise $r_1\,e_1$ drawn perpendicular to $O\,R_2$ extended is the displacement for crank position $O\,R_2$, the construction being precisely as in Fig. 444.

It often makes the diagram clearer to lay off the angular advance below the line of motion also, as shown at $O\,r_2$. *The upper point* r_1, *from which to measure the displacement, is then used for crank positions from* $O\,r_2$ *to* $O\,r_1$ *and the lower point* r_2 *for positions from* $O\,r_1$ *to* $O\,r_2$. In this case, the valve displacement for crank position $O\,R_2$ is $r_2\,e_2$, which it will be seen is the same as $r_1\,e_1$.

1611. Port Opening.—Suppose the valve in Fig. 442 to move to the left. Admission of steam through port S_2 will take place when the valve has moved a distance l equal to the lap, and the port opening will increase until the valve reaches the end of its travel, when *maximum port opening* will occur. This is not necessarily equal to the width of the port, as it is sometimes made less and sometimes greater. The amount that the port is open at any time, however, is evidently equal to the displacement from mid-position to the left minus the lap.

The movement of the valve to the left also opens port S_1 to the exhaust, the amount that it is open being equal to the displacement of the valve to the left minus the inside lap i. In like manner, the opening of S_1 and S_2 to steam and exhaust, respectively, is governed by the laps l_1 and i_1 and the amount of the displacement of the valve to the right.

1612. Diagram for Plain Slide-Valve.—Since, as we have seen, the port opening is equal to the displacement minus the lap, it can always be determined from the displacement diagram previously explained, provided the lap is known. Moreover, as the points of admission, cut-off, compression, and release occur when the port openings to steam

and exhaust are zero, the crank and piston positions for these points can easily be found.

On the following pages are a series of valve diagrams, a sectional view of a slide-valve and ports being placed under each one. Each sectional view is drawn to the scale of the diagram above it and shows the piston and valve positions corresponding to the diagram. In these diagrams, the distance the valve has moved from mid-position was found by the method already explained, and the port opening and, hence, the points of cut-off, compression, etc., by taking account of the laps. As a matter of convenience, the effect of the connecting-rod has been neglected.

1613. In Fig. 446, let $A\ C$ represent the stroke, $A\ B\ C$ the crank circle, and $a\ b\ c$ the eccentric circle. The piston is

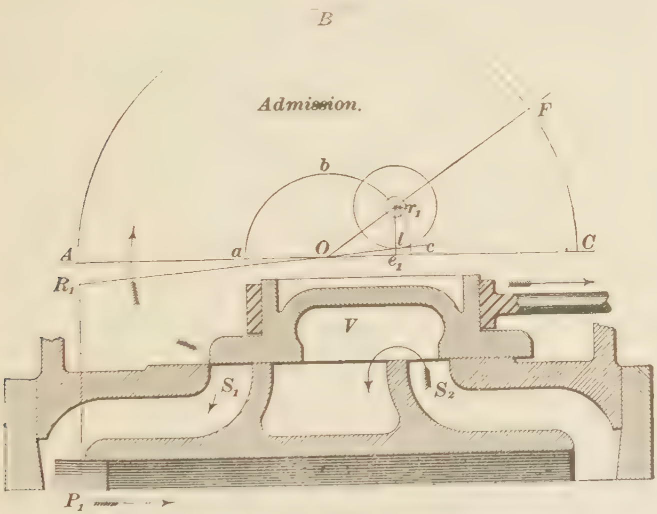


FIG. 446.

supposed to move from left to right, and angle $F\ O\ C$ is the angle of advance.

If the crank position should be $O\ A$, the displacement of the valve would be $r_1\ e_1$, the perpendicular from r_1 to $O\ A$ extended; and if $r_1\ l$, equal to the lap, is laid off on $r_1\ e_1$, $l\ e_1$ will be the port opening, since the port opening equals the

displacement minus the lap. As the port opening at a dead point is the lead, $l c_1$ will equal the lead.

1614. Now, with $r_1 l$ as a radius, a circle is described about r_1, which we will call the lap circle. Also, an inside lap circle is described about the same center with a radius equal to the inside lap. Then, if the crank is in position $O R_1$, so that, when extended, its center line will be tangent to the outside lap circle, the displacement of the valve will be equal to the outside lap, and the valve will be at the point of admission. The sectional view of Fig. 446 shows the valve V in this position, with steam just entering the cylinder through port S_1. In the meantime, steam is being exhausted from the other end of the cylinder through port S_2. The center of the piston is at P_1.

1615. Fig. 447 shows the crank position at $O R_2$, the piston being in the corresponding position at P_2. $O R_2$ is

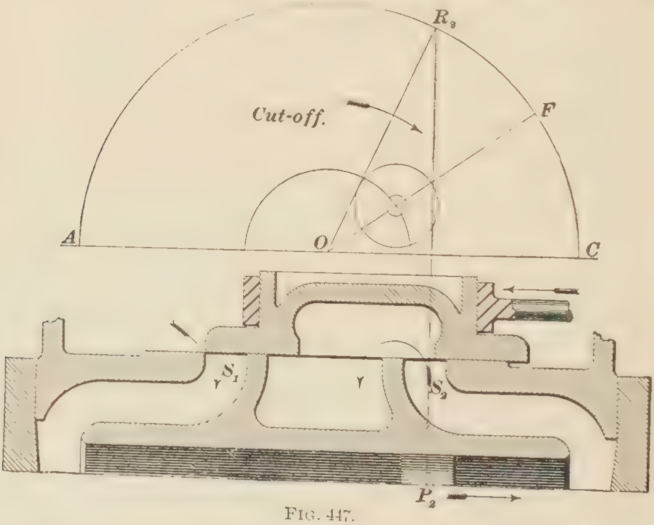


FIG. 447.

tangent to the lap circle, the displacement is again equal to the lap, and steam is cut off from the left end of the cylinder, but continues to exhaust from the right end of the cylinder. The valve is in the same position as before when

it was just opening S_1, but now it is moving in the opposite direction and is just closing the port.

1616. The next event to take place is the closing of

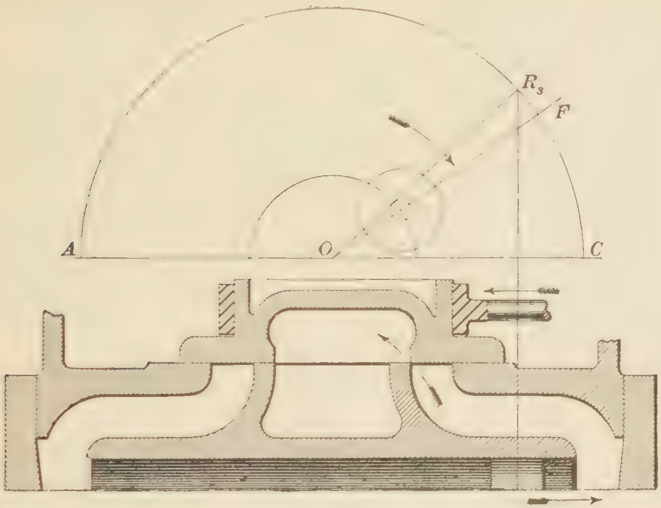


FIG. 448.

port S_2 to the exhaust at the point of compression. As the piston moves to the right and nears the end of its stroke,

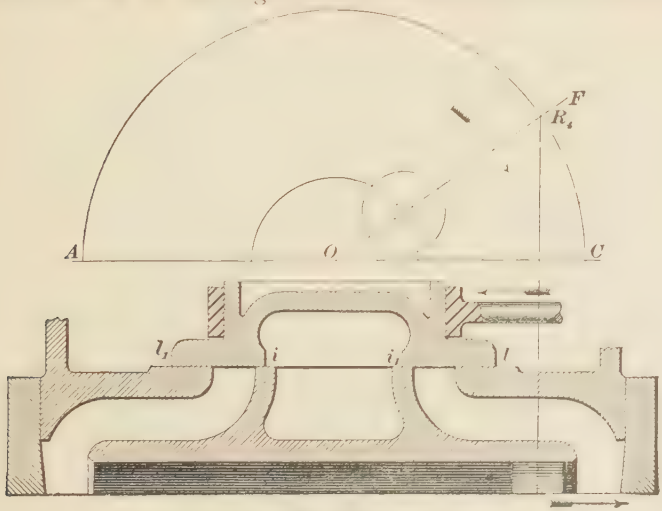


FIG. 449.

the crank reaches position $O\ R_3$, Fig. 448, tangent to the inside lap circle. The displacement is, therefore, equal to the inside lap, the valve is closed, and the steam enclosed in the right end of the cylinder will be compressed during the remainder of the stroke.

1617. In Fig. 449, the crank has reached the line of the angle of advance. The displacement is zero, bringing the valve in mid position as shown. Heretofore, the valve has been displaced to the right of the center line $m\ n$ (Fig. 442) of the exhaust port, and the acting edges of the valve have been l_1 and i_1. Now, the valve is to be displaced to the left, edges i and l are to act, so the plan before referred to of laying off the angle of advance below $A\ C$ has been adopted.*

1618. In Fig. 450, this has been done, and with r_2, the intersection of the angle of advance line $O\ F_2$, and the

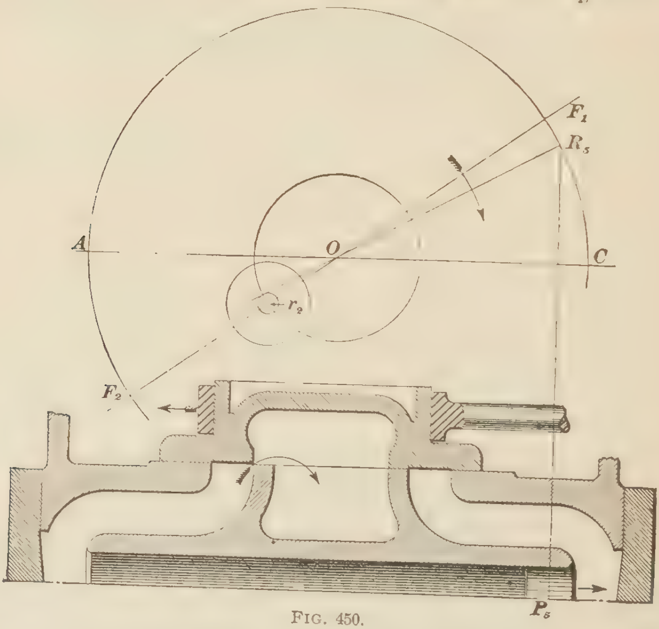


FIG. 450.

* Evidently where the laps on each end are the same it is not necessary to do this, but it is clearer and more consistent to do so.

eccentric circle as a center, the two lap circles were drawn corresponding to laps l and i, which in this case are equal to the other laps. Now, suppose the crank line produced to be $O R_5$, tangent to the inside lap circle. The valve will be at the point of release, and the steam, which has been expanding in the left end of the cylinder, will discharge. The piston is at P_5, very near the end of the stroke, and steam will shortly be admitted to the right side of the piston. Then will follow cut-off, compression, and release, as before, only for the opposite ends of the cylinder.

1619. There is but one other new position to be considered; that is, maximum port opening. Fig. 451 shows

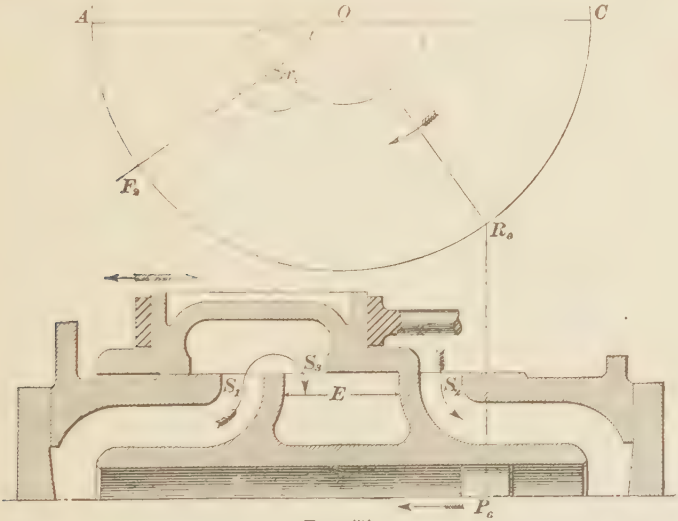


FIG. 451.

the crank at $O R_6$ at right angles to the angle of advance line $O F_2$, and the piston moving to the left on the return stroke. The valve displacement is the perpendicular distance from r_2 to $O R_6$, which is $O r_2$, the greatest it can possibly be, and the port opening is $O t$. From the sectional view, it will be seen that port S_2 is wide open to take steam, and port S_1 is wide open to the exhaust.

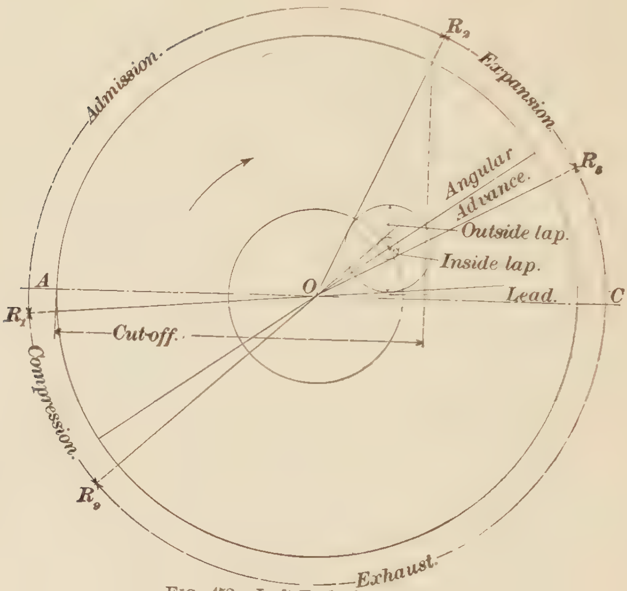


FIG. 452. Left End of Cylinder.

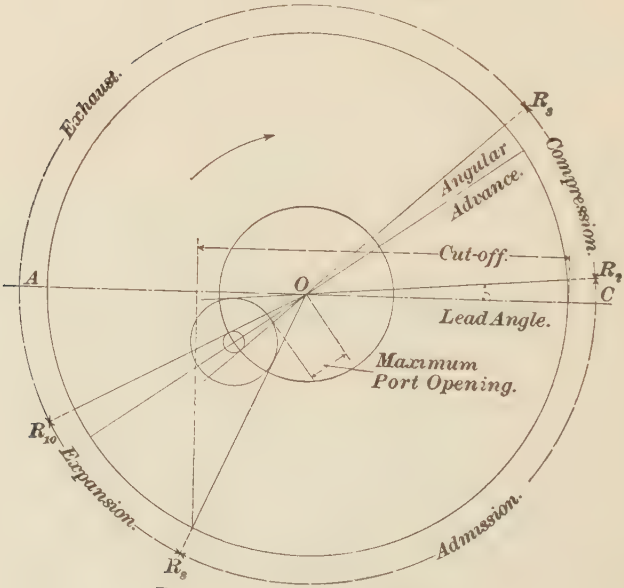


FIG. 453. Right End of Cylinder.

1620. Separate Diagram for Each End of the Cylinder.—Figures 452 and 453 show these various diagrams combined. To make them clearer, the events that take place in the left end of the cylinder during one revolution are represented in Fig. 452, and those that occur in the right end, in Fig. 453. In Fig. 452, admission begins at crank position $O\,R_1$, cut-off takes place at $O\,R_2$, release at $O\,R_5$, and compression begins at $O\,R_9$. In Fig. 453, for the other end of the cylinder, these four events occur at $O\,R_7$, $O\,R_8$, $O\,R_{10}$, and $O\,R_3$.

SLIDE-VALVE DESIGN AND PROBLEMS.

1621. In designing a slide valve, the sizes of the steam ports and the maximum port opening must be calculated and the width of the bridge determined. By **bridge** is meant the wall between the steam and exhaust ports. Then, when the lead and points of cut-off and release or compression have been decided upon, the laps, travel, and angle of advance can be found by means of the diagram, and the width of the exhaust port can be calculated. Sometimes the travel of the valve is fixed at first, leaving the laps and angle of advance to be found.

1622. Width of Steam Port.—In a case like that of a slide-valve engine, where the same passages are used both for admitting and exhausting steam, the steam ports should be designed to allow a free exhaust with the smallest possible back pressure. Evidently the high-pressure steam from the boiler does not require as large passages as the low-pressure exhaust steam, so that satisfactory action can be obtained by giving a port opening during admission less than the full width of the port, but wide enough to prevent a reduction in the pressure of the steam, called **wire-drawing.** Experiments show that a mean steam velocity of 100 feet per second, or 6,000 feet per minute, will give a good exhaust, and that a port opening of from .6 to .9 the width of the port, according to the conditions, will give a free admission.

The width of the port for a given area depends, of course, upon its length, which should be made equal to the diameter of the cylinder, or somewhat less.

EXAMPLE.—Given, diameter of cylinder, 12 inches; piston speed, 600 feet per minute; length of ports, 10 inches. What should be the width of the ports and port opening?

SOLUTION.—With a mean steam flow of 6,000 feet per minute, we would have the area of the port in square inches $\times$ 6,000 = the area of the piston in square inches $\times$ the piston speed in feet per minute, or,

$$\text{area of port} = \frac{\text{area of piston} \times \text{piston speed}}{6{,}000}.$$

$$\text{Here, } A = \frac{12^2 \times .7854 \times 600}{6{,}000} = 11.3 \text{ square inches.}$$

This, divided by 10, gives a width of port of $1\frac{1}{8}$ inches, nearly. Ans.

For the width of the port opening, we have $1\frac{1}{8} \times .9 = 1$ inch, nearly. Ans.

1623. Width of Exhaust Port.—The exhaust port must be wide enough to prevent a reduction of its area to less than the area of the steam ports when the valve is at its maximum displacement. In Fig. 451, where the valve is in that position, the opening S_3 into the exhaust port is equal to the width of the steam port S_1 or S_2. The total width E is equal to S_3, plus the distance the inside edge of the valve has traveled from mid-position, after taking out the distance it has traveled over the bridge. Hence, the

Rule.—*To find the width of the exhaust port, add together the width of the steam port, half the travel of the valve, and the inside lap; from their sum, subtract the width of the bridge.*

When designing a valve, the travel and inside lap are usually two quantities to be determined, making it necessary to leave the calculation of the width of the exhaust port until the last.

1624. Width of Bridge.—The width of the bridges between the steam and exhaust ports is generally made about equal to the thickness of the cylinder wall, so that the shrinking will be equal when the casting cools in the

mold. In any case, they must be wide enough, so that the outside edges of the valve will not uncover the exhaust port. From Fig. 442, it is clear that the maximum displacement one way must not be greater than the width of the bridge + the width of the steam port + the outside lap, $= b + s + l$; it should be $\frac{1}{4}$ inch less than this in medium-sized engines to insure a steam-tight joint when the valve is at maximum displacement.

1625. Amount of Lead.—In general, it can be stated that the lead in stationary engines varies from zero to $\frac{1}{4}$ inch, and is generally not far from $\frac{1}{16}$ inch. No *rule* can be given, however. The amount of lead must be determined for each particular case, sometimes by experiment after the engine is erected.

Lead serves to give the piston full steam pressure at the beginning of the stroke. The tendency of a small lead is to cause the piston to move under a reduced pressure through part of the stroke, especially when the ports are small and the clearance space large. On the other hand, little or no lead gives good results with some engines where the compression is sufficient to produce a pressure at the beginning of the stroke nearly or quite equal to boiler pressure. A quick-acting valve requires less lead than one opening slowly.

1626. Point of Cut-Off.—By turning to Figs. 452 and 453, it will be perceived that an early cut-off on a plain slide-valve engine necessitates an early compression, which becomes excessive when the cut-off takes place before about $\frac{2}{3}$ stroke. Hence, a plain slide-valve is seldom arranged to cut off earlier than $\frac{2}{3}$ or $\frac{3}{4}$ stroke, except in the case of high-speed engines and locomotives, where the compression is not so objectionable and, indeed, is often an advantage.

1627. General Problems.—Given, stroke of an engine, 18 inches; length of connecting-rod, 45 inches; cut-off, $\frac{2}{3}$ stroke; release, $\frac{11}{12}$ stroke; lead, $\frac{1}{8}$ inch; width of steam ports, $1\frac{1}{8}$ inches; maximum port opening, 1 inch; to

find the inside and outside laps, travel of the valve, the angle of advance, and to draw a section of the ports and valve.

At first it will be convenient to consider the motion as harmonic, neglecting the effect of the connecting-rod. In laying out a diagram, it is ordinarily inconvenient to draw the crank circle full size, and sometimes the eccentric circle is drawn to a reduced scale also. In this case, we will draw the crank circle to a scale of 1 inch = 6 inches, and the valve circle half size. All the measurements and parts of the diagram pertaining to the valve will, of course, be half size also.

1628. From the center O, Fig. 454, upon the stroke line, describe the crank circle $A\ C$. To find the outside lap,

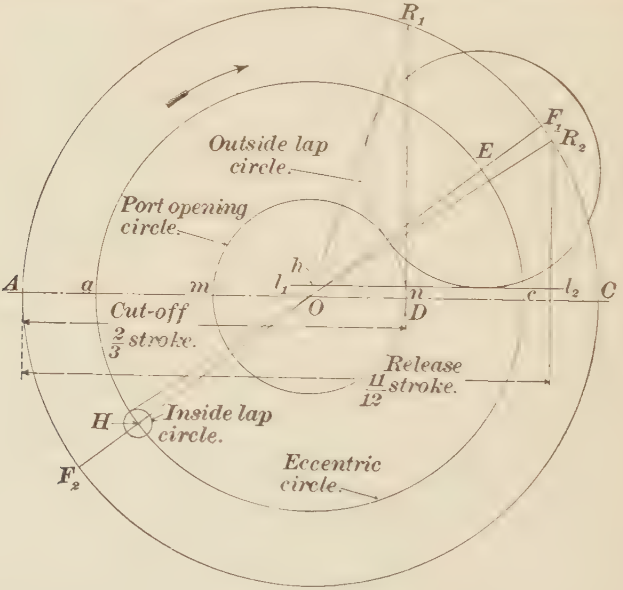


FIG. 454.

we have given the point of cut-off, the maximum port opening, and the lead. Hence, describe the circle $m\ n$ about O with a radius equal to the maximum port opening; draw an indefinite line $l_1\ l_2$ parallel to $A\ C$ and above it a distance

equal to the lead; finally, draw the crank position $O\ R_1$ for cut-off at $\frac{2}{3}$ stroke, R_1 being perpendicularly above D, laid off on the stroke line, as shown in the figure. D happens to fall on the port opening circle in this instance, but does not necessarily do so. Now, it will be evident by reference to Figs. 446, 447, and 451, and the accompanying matter, that a circle drawn tangent to $O\ R_1$, $l_1\ l_2$, and circle $m\ n$ will be the outside lap circle, the radius of which will equal the outside lap of the valve. The center of this circle is found to be at E. It can be located readily by bisecting the angle $R_1\ h\ l_2$; the center must then fall at some point on the bisector.

1629. To determine the valve travel, we have simply to draw the eccentric circle $a\ c$, having the center O, through the point E. The diameter of this circle will be the travel.

1630. We have at once, also, the angle of advance by drawing the line $F_1\ F_2$ through points E and O, making the angle $F_1\ O\ C$ with $A\ C$.

1631. Finally, to obtain the inside lap, draw crank position $O\ R_2$ for $\frac{11}{12}$ of the stroke. $O\ R_2$, being beyond the angle of advance line $O\ F_1$, should properly be produced beyond O. As the laps are equal, this is not necessary, however. The circle with center at H drawn tangent to $O\ R_2$ produced will then be the inside lap. Measuring the diagram, we obtain the following dimensions, nearly: Travel, $4\frac{7}{16}$ inches; outside lap, $1\frac{7}{32}$ inches; inside lap, $\frac{1}{8}$ inch; angle of advance, 37°.

The section of the valve and ports is shown in Fig. 442, which is one-fourth size. To draw the valve, its length, or the distance between the edges of the valve faces, must be known, and is easily determined by first drawing a section of the ports. By the rule previously given, the width of the exhaust port should be $1\frac{1}{8} + 2\frac{3}{16} + \frac{1}{8} - 1 = 2\frac{7}{16}$ inches. It is drawn $2\frac{1}{2}$ inches wide, $1\frac{1}{4}$ inches on each side of the center line $m\ n$. The bridges are drawn 1 inch thick on the assumption that the cylinder walls are of that thickness, and the steam ports are $1\frac{1}{8}$ inches wide. Now, having completed the

section of the ports, the outside laps should be laid off outward from the outside edges of the steam ports and the inside laps from the inside edges. The valve faces will then be determined and the valve section can be completed.

1632. Equalizing the Cut-Off.—This valve is designed to cut off at $\frac{2}{3}$ stroke and to have a lead of $\frac{1}{8}$ inch, without considering the irregularity produced by the connecting-rod. Should an indicator be applied to the completed engine, however, it would show, provided the engine was running "over," that the cut-off occurred *later* than $\frac{2}{3}$ stroke on the forward stroke and *earlier* on the return. In other words, steam would be admitted to the head end for a longer time than to the crank end. One way to overcome this is to give more lap to the end of the valve towards the head end of the cylinder, which will hasten its action during the forward stroke, and to reduce the lap on the other end, in order to retard the action on the return stroke.

1633. Fig. 455 shows a diagram for the valve laid out in this way. $O\,R_1$ and $O\,R_2$ are the crank positions at cut-off, and H is the center of the smaller outside lap circle. The points R_1 and R_2 are determined by laying off the distance $A\,n$, on the line $A\,C$, equal to $\frac{2}{3}\,A\,C$. Then, produce $A\,C$ to the left a distance equal to or greater than the length of the connecting-rod. With a radius equal to the length of the connecting-rod to the same scale as that to which the crank-pin circle was drawn—that is,

$$\frac{45}{6} = 7\tfrac{1}{2}'' \left(= 7\tfrac{1}{2} \div 2 = 3\tfrac{3}{4}'' \text{ in the figure}\right)—$$

and a center on $A\,C$ produced, describe the arc $n\,R_1$, which cuts the crank-pin circle at R_1. R_1 is the position of the crank-pin when the steam is cut off from the head end of the cylinder. Also, lay off $C\,m$ equal to $\frac{2}{3}\,A\,C$, and with the same radius and a center on the line $A\,C$ produced describe the arc $m\,R_2$, intersecting the crank-pin circle at R_2. R_2 is the position of the crank-pin when the steam is cut off from the crank end of the cylinder. Draw $O\,R_1$ and $O\,R_2$. The valve travel remains the same; hence, drawing the eccentric

circle (sometimes called the valve circle), the center of the lap circle must lie on this circle. Consequently, bisecting the arc included between $O R_1$ and $n c$, the point E is obtained, which must be the center of the outside lap circle. Drawing $E H$ through O, it intersects the valve circle in H. Draw $O R_2$, and with H as a center describe a circle which shall be tangent to $O R_2$. This circle is the other outside lap circle.

From this it will be perceived that equalizing the cut-off by varying the laps is done at the expense of the lead, which in this case is nearly $\frac{1}{2}$ inch for the return stroke. It also causes an unequal port opening, which is of minor

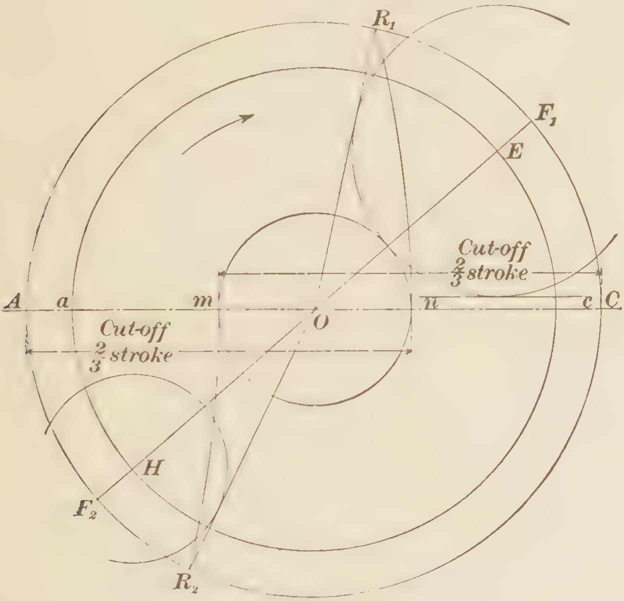


FIG. 455.

importance, however, provided the maximum opening on the end at which it is the least is sufficient.

1634. A better way of equalizing the cut-off is to provide for the use of a rocker which, if properly designed, will neutralize the irregularities due to the connecting-rod

at the points of cut-off, without disturbing the leads. The method of procedure is as follows:

First, determine from the valve diagram the travel, lap, angle of advance, etc., *for one end*, taking account of the connecting-rod. Next, lay out a diagram like Fig. 456, where $A\ C$ is the stroke of the cross-head, $D\ F\ B\ H$ is the crank-pin circle, and $h\ d\ f\ b$ is the valve circle, with a diameter in this case smaller than the travel of the valve, because of the multiplying effect of the rocker. Cut-off is to take place at $\frac{3}{4}$ stroke. Crank positions corresponding are $O\ F$ and $O\ H$, and we will let crank positions at admission be $O\ D$ and $O\ B$.

When a direct-acting rocker is used, the eccentric must be 90° + the angle of advance ahead of the crank. In the figure, eccentric positions $O\ f$ and $O\ h$ are laid off 90° + the angle of advance (as found from the diagram) ahead of crank positions $O\ F$ and $O\ H$. In like manner, eccentric

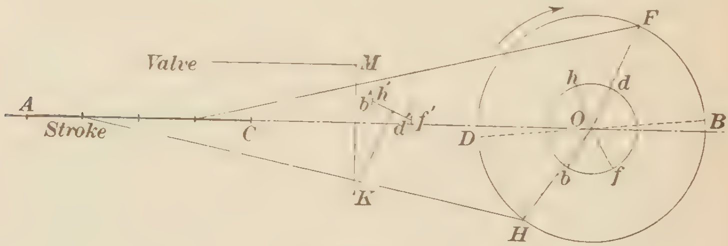


FIG. 456.

positions $O\ d$ and $O\ b$ are drawn corresponding to the points of admission.

Now, we know that a slide-valve is in the same position at both cut-off and admission. Hence, with a radius equal to the length of the eccentric-rod, and with admission and cut-off points d and f as centers, strike arcs d' and f'. The point of their intersection will be the point at which the eccentric-rod pin should be at admission and cut-off on the forwards stroke. In like manner, the intersection of arcs b' and h' gives the point for the return stroke. Connecting these points and erecting a perpendicular halfway between them, we have the central position of one arm of the rocker.

The other arm $K M$ should be perpendicular to the valve stem at the central position, and the point of intersection K must be chosen so as to make the two lever arms proportional to the throw of the eccentric and the travel of the valve, respectively.

By methods similar to the foregoing, the release, or compression, may be equalized.

1635. Whenever, for any purpose, a rocker is used that either increases or diminishes the action of the eccentric on the valve, the valve's travel is to be used instead of the throw of the eccentric for all calculations and constructions connected with the valve diagram, as lap, lead, or cut-off.

1636. Modifications of the Slide Valve.—There are many modified forms of the slide valve, but the valve diagram can be applied to any of them that are operated by an eccentric. One of the most common modifications is the piston valve, which is illustrated in connection with the triple-expansion marine engine in Art. **1309.** When a piston valve takes steam at the ends and the exhaust steam passes out through the center, its action is in every respect like that of the plain slide valve. Sometimes, however, steam is admitted to the ports through the central part of the valve, and steam is exhausted at the ends. The steam lap, therefore, is on the inside and the exhaust lap at the ends of the valve, and the only difference is that the valve must move in the opposite direction to that in which it would move if it were a plain slide valve, and the eccentric would have to be moved around the shaft 180°.

1637. A **Trick valve** is shown in Fig. 457. It was designed to give a quick and full opening of the port with a small travel of the valve.

In Fig. 457, A shows the valve in mid-position, while B and C show the valve in two other positions. As will be seen, the valve is hollow, having a passage way H through it; otherwise, the valve corresponds very closely to the ordinary **D** valve before described. A movement to the right

of a distance equal to m will bring the edge p of the valve to the edge of the port S, as shown at B, so that any further movement to the right will admit steam to the cylinder. But this same movement has brought the edge f of the passage in line with the edge g, and any further movement to the right will admit steam to the passage, and, hence, to the left-hand port S, from beneath the valve (see C, Fig. 457). Suppose the valve to move, say $\frac{1}{16}''$ to the right from its position at B; then, the edge f will be $\frac{1}{16}''$ beyond the edge g, and edge p will be $\frac{1}{16}''$ beyond the outer edge of port S. This shows that a movement of the valve which would ordinarily have opened the port $\frac{1}{16}''$ had a **D** valve been used,

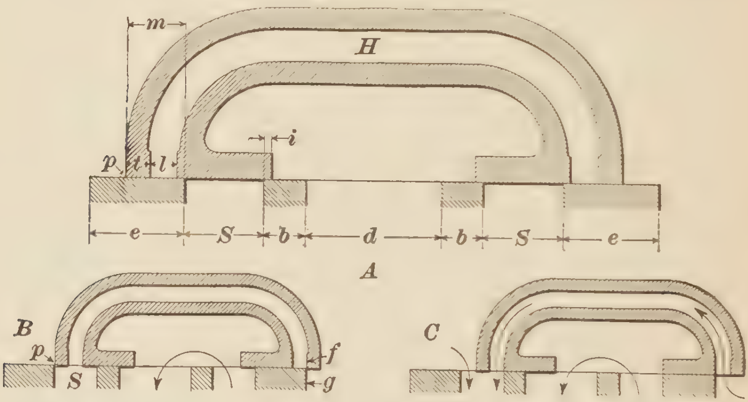


FIG. 457.

has opened the valve twice $\frac{1}{16}''$ or $\frac{1}{8}''$. C, Fig. 457, shows the valve in its extreme position to the right, giving full port opening to the steam and exhaust. The inside lap is shown by i, and the outside lap really equals m.

1638. The parts of the valve should have the following dimensions: $l = \frac{1}{2}(S - t)$; half travel $= S + i$; $e = 2m - t$. The width of the exhaust port d should be equal to $S + m + i + l - b$. The diagram is drawn as for a simple valve, remembering that the width of the opening to exhaust is S and of the opening to steam $2\,l$. In marine engines having a large diameter and short stroke, double-ported valves, as

shown in Fig. 458, are often used to obtain a sufficient port opening with a small travel. It will be seen to consist of two **D** valves, each with its ports and laps. Steam surrounds the outer valve and also the inner one, entering through B at the sides. Here, again, the diagram can be applied, the actual or total port opening being twice that of either valve considered separately.

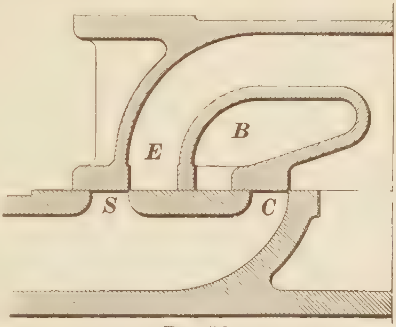


FIG. 458.

TO SET THE SLIDE VALVE.

1639. Principles Involved.—Setting a valve is more a matter of common sense than of rule. If the principles are understood there should be no difficulty.

After the valve and connections are constructed, there are usually two available means of adjustment—the valve spindle can be lengthened or shortened, and the eccentric can be moved about the shaft.

Lengthening or shortening the valve spindle serves to make the valve travel equally each way from mid-position. For example, if the valve travels an inch too far towards the head end, shortening the spindle by half that amount will pull the valve one-half inch towards the crank end and cause its displacement to be equal each way. Moving the eccentric simply hastens or retards the action of the valve, according as it is moved ahead or back; it alters the angle of advance.

1640. To Put the Engine on Dead Center.—Valve setting frequently involves setting the crank on the dead center, which may be accurately done as follows: Make a center-punch mark on the frame of the engine near the turned part of the rim of the fly-wheel. Place the crank a short distance from one dead point, and with the punch

mark as a center, describe an arc on the wheel rim with a tram, which is simply a steel rod with its ends bent at right angles and sharpened. Also, scratch a mark across the cross-head and guide with a scriber. Turn the engine past the center until the mark on the cross-head corresponds again with the line on the guide, and make another mark on the rim with the tram. With the center of the fly-wheel as a center, describe a circular arc on the fly-wheel rim which will intersect the two short arcs just described. Now, bisect the arc included between the points of intersection, and turn the wheel until this last point is at a distance from the punch mark on the frame equal to the length of the tram. The engine will then be on one center and it can be set on the other center in the same way.

Another less accurate but simpler way is to turn the engine over slowly through one revolution and to follow the cross-head with a scriber held in the hand. Just as the point of reversal is reached a line can be scratched on the guide, marking one center.

1641. To Set the Valve for Equal Lead.—It is first necessary to make the valve move centrally by adjusting the valve spindle and then to make it act at the right time by moving the eccentric.

Set the crank on a dead point, and give the eccentric the proper angular advance as near as can be judged. Measure the lead. Set the crank on the other dead point and again measure the lead. Then, move the valve on the spindle half the difference of the two leads, and finally give the valve the right lead by moving the eccentric. The lead should then come the same at the other end.

1642. A Second Method.—This method is convenient when it is difficult to turn an engine by hand. Loosen the eccentric and turn it around on the shaft to give maximum port opening at first one end and then the other. Make the openings equal by changing the length of the valve spindle by half their difference. Then, set the engine on a dead point and give the valve the proper lead by the eccentric.

SHIFTING ECCENTRICS.

1643. It is very common to regulate the speed of slide-valve engines by means of a throttling governor which varies the pressure of the steam before it enters the cylinder. Steam is often admitted to the cylinder without throttling, however, and the speed of the engine is then regulated by varying the time during which the steam is admitted; that is, by varying the point of cut-off. Slide-valve engines are often governed in this way, especially when made to run at high speed, the regulation being accomplished by shifting either the position or the throw of the eccentric.

1644. Changing the Angle of Advance.—One way of doing this is to have the eccentric loose on the shaft and connected with a governor in such a way that it will be rotated back and forth with the fluctuations of speed, thus changing the angle of advance. The effect of changing the angle of advance is shown in Fig. 459. Let $O F_1$ be one position of the eccentric. Cut-off occurs at crank position

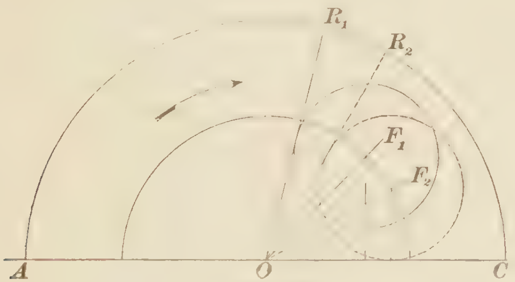


FIG. 459.

$O R_1$. Now, move the eccentric back to $O F_2$, which will decrease the angle of advance and bring the cut-off later in the stroke at $O R_2$. It will be seen, however, that the lead also varies by a large amount, which is not always desirable.

1645. Changing the Eccentricity.—Another way by which the cut-off can be changed is by varying the throw of the eccentric, leaving the angle of advance unchanged. Suppose, in Fig. 460, the eccentricity to be changed from

$O\,r_1$ to $O\,r_2$. The cut-off will change from crank position $O\,R_1$ to $O\,R_2$. The lead will also change, but in this case

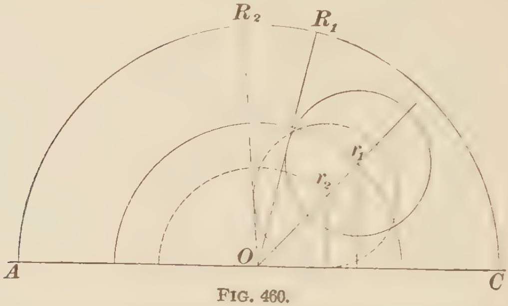


FIG. 460.

the later the cut-off the greater the lead, whereas in the other case where the angle of advance was changed the lead decreased as the cut-off became later.

1646. Changing Both Angle of Advance and Eccentricity.—A combination of the two methods, by diminishing the angle of advance and increasing the eccentricity, or increasing the angle of advance and decreasing the eccentricity at the same time, may be made to give a variable cut-off with constant lead. In Fig. 461 two positions of the eccentric are shown at $O\,F_1$ and $O\,F_2$. In the

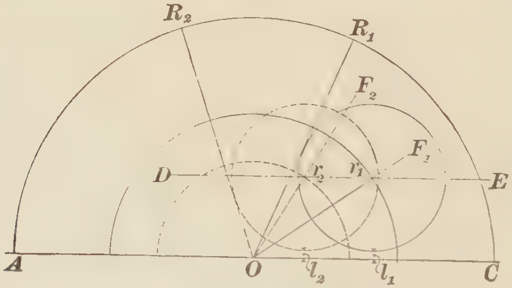


FIG. 461.

former position the eccentricity is $O\,r_1$, and in the latter $O\,r_2$, r_1 and r_2 being in the line $D\,E$, parallel to $A\,C$, which brings the lead l_1, equal to the lead l_2. The eccentric, therefore, has to shift across the shaft on the line $D\,E$, as indicated in Fig. 461.

1647. Shifting Eccentric with Variable Lead.—Fig. 462 shows in principle another method sometimes used that gives a variable lead to the valve. *A C* is a collar keyed to the shaft with lugs for bolts. The eccentric is slotted for the shaft and bolt *b*, the latter serving to clamp the eccentric which swivels on the stud *s*.

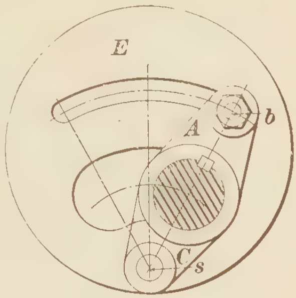


FIG. 462.

Many devices are used to vary the cut-off in some one of the foregoing ways by the action of the governor, some of which will be taken up with Shaft Governors.

DOUBLE VALVE GEARS.

1648. We have seen that an early cut-off with a plain slide valve is accompanied by an excessive compression and early release. To take advantage of the superior economy incident to the use of an earlier cut-off, and at the same time to avoid too early release and compression, double valves are used.

A double valve consists of a main valve, which is set to give the proper lead and compression, or exhaust, and a separate and independent cut-off valve. The latter is sometimes placed in a separate steam chest communicating with the steam chest of the main valve through ports under the cut-off valve. By cutting off the admission of steam to the main valve chest, steam is prevented from entering the cylinder whether the main valve is open for steam or not. A better and more common way, however, is to place the cut-off valve directly on the back of the main valve, which then acts as a valve seat for it. When this arrangement is used the cut-off valve is generally made in two parts, which may be separated or brought together by means of a right and left hand screw, thus varying the lap, and, hence, the cut-off. This arrangement is known as the Meyer valve.

1649. Meyer Valve.—In Fig. 463 a section of such a valve is shown. A is the main valve, which has two passageways C and D, the part between the passageways being an ordinary D valve. On the back of the main valve is the cut-off valve, consisting of two flat plates B, B, con-

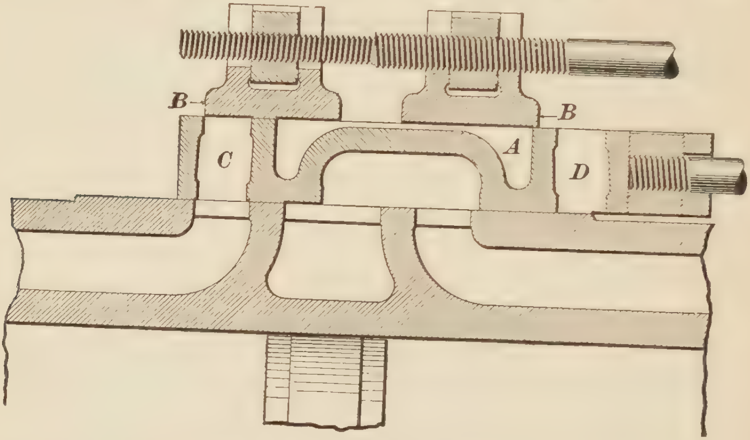


FIG. 463.

nected by a right and left hand screw. The main and cut-off valves are moved by separate eccentrics. A conventional way of representing a Meyer valve is shown in Fig. 464 where both parts are in mid-position—positions they can

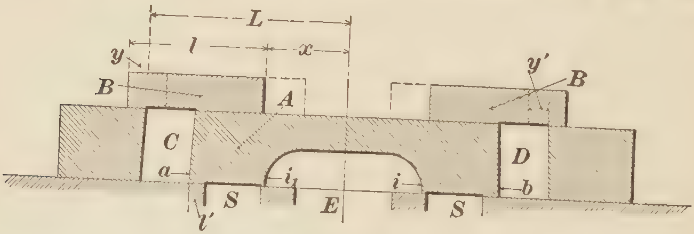


FIG. 464.

not occupy at the same time when connected with their eccentrics.

The cut-off valve acts solely to cut off steam, steam being admitted by the edges a and b of the main valve, Fig. 464. Cut-off is effected through the closing of the passages C and

D by the cut-off valve. It is evident, however, that the steam ports S, S in the cylinder will be closed by the edges a and b of the main valve at some point of its travel, so that if the cut-off valve is to serve its purpose it must act before the edges a and b cut off. Hence, to allow a wide range of cut-off, as well as to avoid excessive compression and a very early release, both of which are controlled by the main valve, the main valve should have a late cut-off.

THE CORLISS GEAR.

1650. A **Corliss gear** can not be laid out from the diagram like a plain slide valve. The dimensions of the parts must be proportioned by trial; but there are certain requirements that must be fulfilled, and these will be explained.

I.—The steam port should be opened rapidly, so as to avoid throttling the steam. This action is aided by the position of the eccentric which, in the Corliss engine, has only a slight angular advance, being nearly at 90° with the crank. The eccentric is, therefore, near mid-position at the points of admission and causes a prompt admission. To obtain the full benefit of this, the rods E and E' (Fig. 282, Art. **1287**) should be attached to the wrist-plate so that the movement of the attaching pin will be symmetrical on each side of the center of the wrist-plate A.

Following out the motion of the eccentric, it will be seen that, as it moves from mid-position, the valve will move less rapidly until, when fully open, it will be nearly at rest. It is not desirable, however, that the rapidity of motion should reduce too quickly; otherwise, as the piston nears mid-stroke, and, consequently, moves more quickly, some wire-drawing of the steam might occur. This is avoided by the arrangement of the rods and levers connecting the valves with the wrist-plate, the principle being similar to that of the slow-motion mechanism described in Art. **1444.**

Fig. 465 is a skeleton diagram of the wrist-plate and connections as found on one make of Corliss engines. The

center of the wrist-plate is O, and the center of the admission and cut-off valve for one end is at C. The parts of the diagram have the same letters as the corresponding parts of the diagram in Fig. 355, Art. **1448.** By plotting the motion, it will be seen that when the wrist-plate has moved through half its motion, or point b has reached point i, the end of the valve lever has been moved only through the arc $c\ i'$, while for the other half of the wrist-plate motion it moves through the greater arc $i'd$.

The action of the lever which connects with the exhaust valve C' is similar.

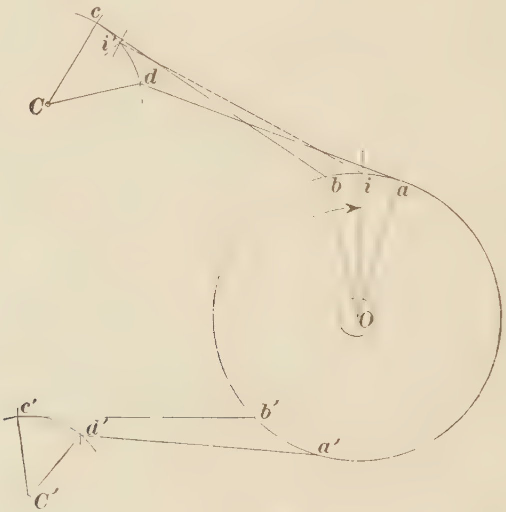


FIG. 465.

II.—The amount of travel of the wrist-plate should be great enough to cause the valve to give full port opening and over-travel from $\frac{1}{4}$ inch to $\frac{3}{8}$ inch besides. For, suppose the valve gave only full port opening for the extreme position of the eccentric; then, for short cut-offs, the valve would be tripped by the governor before the port was opened wide, and for later cut-offs the valve would move too slow, owing to the retarding action of the eccentric spoken of above. With over-travel this retarding action does not become serious until *after* the port is wide open.

III.—Having decided upon the valve travel, the lever arms and connections can be proportioned, and the next point to be considered is the angle which the lever arm *C c* in Fig. 465 should make with the rod *c b* connecting with the wrist-plate. It is clear that the greater this angle the more rapid the valve motion, but it should not be great enough to permit of there being any danger of the lever arm passing the center when in position *C d*.

1651. The Lap of the Valves.—The question of what lap to give the valves of a Corliss engine is somewhat complex.

A little consideration will show that cut-off by the action of the governor can occur only while the wrist-plate is moving one way. If the valve is not tripped before the motion of the wrist-plate reverses, the cut-off is given by the lap of the valve near the end of the stroke. If, therefore, the eccentric was set at right angles to the crank, its motion one way would continue until the piston reached about half stroke, so that cut-off could occur up to half stroke. Moreover, with the eccentric in this position, the valve would have the most rapid movement possible when opening.

But the eccentric can be at right angles to the crank only when there is no lap, and the greater the lap the more the eccentric must be set ahead to give the steam valves the proper lead. With the eccentric set ahead it will have reached the position of its greatest displacement either way *before* the piston reaches half stroke, the range of cut-off will be shortened, and the movement of the valve made less rapid.

It would, therefore, be advisable that the eccentric have no angular advance, and the valves no lap other than that necessary to make a steam-tight joint when closed, were it not for the exhaust valves. These valves are operated by the same eccentric as the admission valves, and could have no lap if the eccentric should have no angular advance.

To obtain a good steam distribution, however, both release and compression should occur *before* the end of the

stroke; with no lap they would take place *just at* the end of the stroke. The only way to bring about these events earlier is to add lap to the exhaust valves, and give the eccentric such angular advance as is necessary to bring about an early action. The reason why this is so should be clear from what has been stated before regarding the slide valve.

Since the eccentric is moved ahead, the admission valves must also have lap, and thus it will be seen that the whole question of lap on Corliss valves is that of making the exhaust valves act properly without interfering with the action of the steam valves. Separate eccentrics for the steam and exhaust valves will allow each to be set as it should be, but usually only one eccentric is used.

THE STEPHENSON LINK MOTION.

1652. In Fig. 466 is shown the **Stephenson link motion** as applied to locomotives. It differs from that shown in Fig. 316, Art. **1330,** in that a rocker *R* is used to actuate the valve, and that the link is suspended from above instead of being supported from below. The principle of the motion is the same, whatever the type of the engine to which it is applied.

In the figure *O* is the center of the driver axle. The two eccentrics *F* and *B* are keyed to the axle, and connect with the slotted link *L* through the eccentric rods *E* and *G*. The slide valve is attached by its stem *V* to the upper arm of the rocker *R*. The rocker in turn is connected with the link through the block *K* and the rocker pin *d*, the former being free to slide in the slot of the link.

The eccentric *F* is set to give the forward movement to the engine and the eccentric *B* to give a backward movement. The raising or lowering of the link is accomplished through the hanger *H* attached to the lower lever on the tumbling shaft *T*. The link itself is suspended from its saddle *S* by the hanger *H*, and the upper lever on the tumbling shaft connects with the reversing lever, not shown, through the reach rod *W*.

When the reversing lever is thrown forward, the link is lowered and the engine will run ahead; when thrown back, the link is raised and the engine will run backwards. In Fig. 466 the link is shown lowered so that the rocker pin *d* comes in line with the eccentric-rod *E*. If the link should be raised to the other extreme, *d* would be in line with eccentric-rod *G*. In either case the link is said to be in **full gear.** Should the rocker pin *d* be at the middle point of the link, the latter would be in **mid-gear.** If the link is

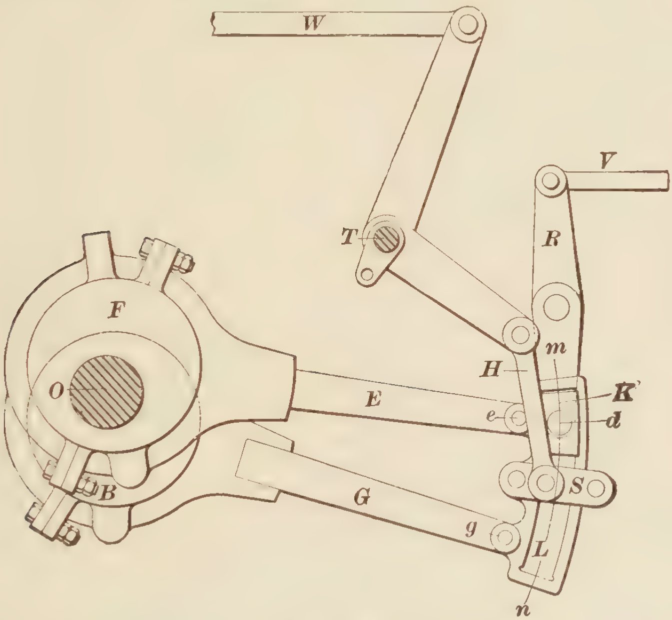


FIG. 466.

in full gear, so that the forward eccentric actuates the valve, it is said to be in *full gear forwards*, and when the backward eccentric actuates the valve, in *full gear backwards.*

1653. It will be remembered that when a reversing rocker intervenes between the eccentric and valve, the crank leads the eccentric by 90° minus the angle of advance of the eccentric. In Fig. 466, therefore, the crank falls within

the angle made by the two eccentrics, and, with the link lowered, *leads* eccentric *F*. If no rocker were used the effect would be to turn the crank around 180°, and it would *follow* eccentric *F*.

In what follows the link motion will be illustrated by skeleton diagrams. The link will be represented by an arc drawn through the curved slot, called the **link arc,** and each eccentric-rod will be represented by a line corresponding to the center line in Fig. 466, drawn from the center of the eccentric *F* to the point *d*. The eccentrics will be represented by straight lines connecting the center of the eccentrics with the center of the axle or shaft.

1654. Action of the Link Motion on the Valve.—While the link motion was designed primarily for reversing, it is found to be well adapted for a variable cut-off gear. Its action will be understood by reference to Figs. 467 and 468.

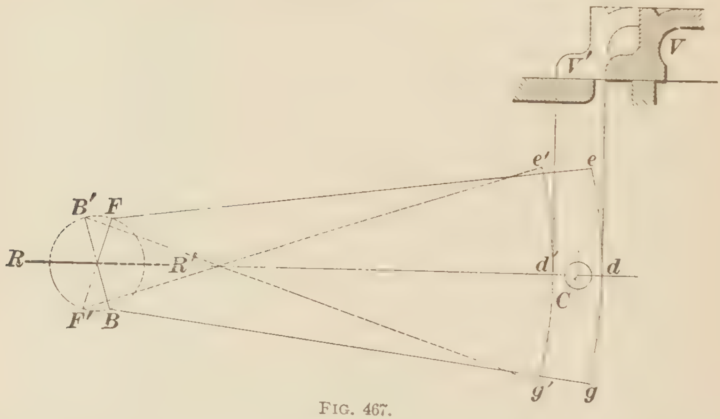


FIG. 467.

In Fig. 467 the solid lines represent the positions of the parts when the crank is on a center and the eccentrics are turned *towards* the link; the dotted lines show the positions when the eccentrics are on the side *opposite* to the link. For convenience, the valve is drawn directly above the link, and it is assumed that no rocker is used. When the crank is on the center at *R*, the link is at *eg*, and the point *d* is displaced a distance *Cd* to the right of the central position *C*. The

valve is also displaced a like amount, as shown at V. As the crank is on a center, the port is open an amount equal to the lead, and the displacement of the valve is equal to the lap plus the lead. In a similar manner, when the crank is at R', the eccentrics F and B will be at F' and B', respectively, and the link will be at e' g', the valve then being displaced to the left a distance $C d'$ equal to the lap plus the lead, as shown at V'.

Now, in mid-gear this is the greatest possible displacement of the valve, its total travel being equal only to the distance $d d'$, or twice the lap plus twice the lead. This will

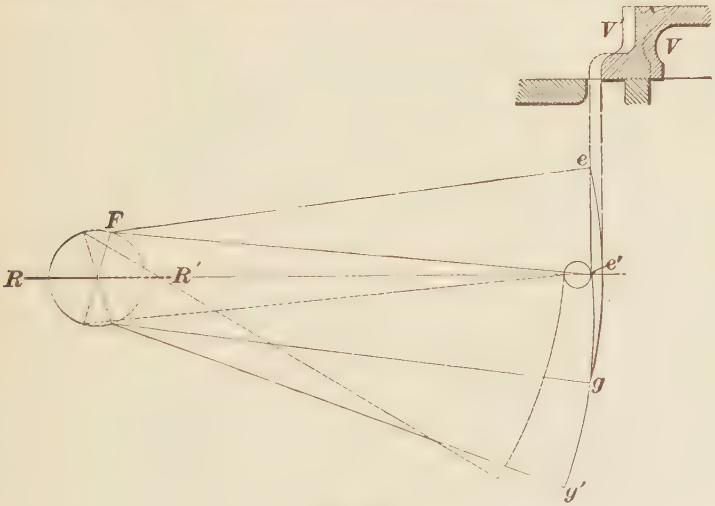


FIG. 468.

be evident by supposing the crank to be at R and to start to rotate in a right-handed direction. The tendency of the eccentric F will be to move point e of the link to the right, thus increasing the valve travel; but eccentric B will have a greater tendency to move the point g to the left, since it is approaching a position at right angles with the eccentric-rod, while eccentric F is moving away from such a position. From what we know of the action of the crank and connecting-rod, therefore, it is clear that B will have a greater influence on the movement of the link than F, and point d will

move to the left. *From this it follows that the greatest port opening in mid-gear is the lead, and that cut-off occurs very early in the stroke.*

1655. Fig. 468 shows the link in mid-gear at $e\,g$, and in full gear forwards at $e'\,g'$. In the latter position the valve is actuated almost entirely by the eccentric F, as in an ordinary slide-valve engine.

The travel of the valve, therefore, is now equal to the throw of the eccentric instead of twice the lap plus twice the lead, as before. Full port opening is now obtained, and cut-off occurs much later.

1656. Now, suppose the crank to be set on the dead center, as in Fig. 468, and the link to be shifted from full gear to mid-gear, or from $e'\,g'$ to $e\,g$. The lead, it will be seen from the two positions V and V' of the valve, increases from a very small amount to the large lead obtained in mid-gear.

The above explanation of the valve's action may be summarized as follows:

In full gear the valve is under the control of one eccentric, which gives it a motion like that given to a plain slide valve by a single eccentric.

In passing from full gear to mid-gear, cut-off becomes earlier, and, hence, compression greater, the travel of the valve diminishes, which makes the port opening less; the lead increases.

In mid-gear the travel of the valve is equal to 2(lap + lead), and the maximum port opening is equal to the mid-gear lead.

1657. Opened and Crossed Rods.—As shown in the preceding figures, the rods are said to be **open.** If the eccentric-rods E and G, in Fig. 466, should be disconnected from the link and rod, and E should be bolted to the lower end of the link, and rod G to the upper end, we should have the arrangement shown in Fig. 469; in this case the rods are said to be **crossed.** The terms open and crossed are given according to the position of the rods *when the eccentrics are turned towards the link.* Thus, in Fig. 467, although

the rods are crossed when the eccentrics point *away* from the link, they are still called open links, because when turned *towards* the link they are not crossed. The action of crossed rods is different from open rods in that the lead *decreases* from full to mid-gear, in which position the motion usually gives no lead. With crossed rods the engine may be stopped by placing the link in mid-gear. This can not be done with open rods, where there is always a small port opening in mid-gear, unless the resistance to be overcome by the engine is so great that enough steam can not be admitted in full gear to run the engine.

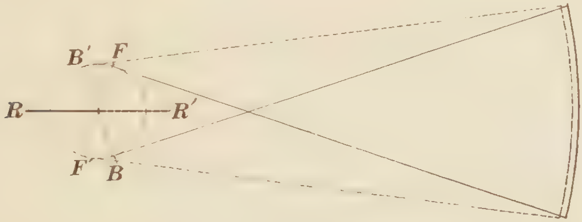


FIG. 469.

Open rods are mostly used, and as the principle of crossed rods is essentially the same, open rods alone will be considered.

DESIGNING THE LINK MOTION.

1658. The Valve and Ports.—The valve should be designed to meet the requirements when the link is in full gear. As in this position it is practically under the control of one eccentric, the valve diagram may be used.

Generally, the widths of the ports, the latest point of cut-off, the maximum port opening, and the full gear lead are known, or assumed, leaving the lap, travel, and angular advance to be found, the latter being the same for both eccentrics. When the link is to be used for reversing only, as in the case of hoisting engines, this part of the process is in all respects like designing a slide valve for an engine without the link motion. A hoisting engine is always run with the link in full gear, and the valve is designed to give the best results in this position.

If the link is to be used for varying the cut-off, however, as well as for reversing, as in the case of the locomotive, the valve should be given enough travel, so that it will not only open the port its full width in full gear, but will *over-travel* or will go beyond the inner edge of the port a certain amount, thus making what has heretofore been called the "port opening" greater than the width of the port. If this were not done the port opening would become so small for the intermediate points between full and mid-gear at which the engine would usually be run as to cause serious wire-drawing of the steam.

1659. As a basis to work upon, proportions of the valve and ports are here given, taken from locomotive practice. They will be convenient for reference in designing a link motion for any type of engine.

The following are average values: Latest point of cut-off, $\frac{7}{8}$ to .9 stroke; lead in full gear $\frac{1}{16}''$ to [illegible]; lead in mid-gear, $\frac{3}{8}''$; inside lap, 0 to $\frac{1}{8}''$.

The accompanying table gives dimensions in inches from locomotives having cylinders over 15″ in diameter:

Width of Steam Port.	Width of Exhaust Port.	Outside Lap.	Valve Travel.
$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	5
$1\frac{1}{4}$	3	$\frac{7}{8}$	$5\frac{1}{2}$
$1\frac{1}{4}$	$2\frac{1}{4}$	1	$5\frac{1}{4}$
$1\frac{1}{4}$	$2\frac{3}{4}$	$\frac{3}{4}$	$5\frac{1}{2}$
$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	$5\frac{1}{2}$

The maximum port opening varied in these cases from $1\frac{5}{8}''$ to 2″.

1660. The Suspension of the Link.—As the eccentrics revolve, the link has a continued vibratory motion about its point of suspension and a swinging motion due to the vibration of its hanger. The result is more or less vertical motion, according to the point at which the hanger is

attached to the link, which, if excessive, will cause undue wear between the link and the block (*K* in Fig. 466). This motion between the block and the link is called the **slip.**

Ordinarily, the link is suspended at one of three points —the center of the link arc, the center of its chord, or at the lower end of the link. When at the first point, the slip is slight in mid-gear, and increases both ways; at the second, the slip increases from mid-gear, but is always *greater* than in the former case; while, at the third point, the action is good when the block is in the lower half of the link, but when in the upper half the slip is excessive. When the engine is to be run with the block at some one point in the link nearly all the time, the latter should be suspended from that point to reduce the slip as much as possible.

Both the hanger which supports the link and the lever to which its upper end is attached should be as long as possible. The object should be to so suspend the link that the saddle pin will always move approximately parallel to the center line of motion.

1661. Proportions of the Link.—The length of the link should not be less than $2\frac{1}{2}$ to 3 times the throw of the eccentric; if less, it is difficult to reverse the engine when the piston is near the end of the stroke, as the link then makes an obtuse angle with the valve stem. By the length of the link is meant the length *l* (Fig. 470) of the chord connecting centers of the link block when it is in the two extreme positions.

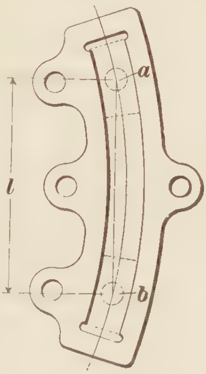


FIG. 470.

The radius of the link is generally made equal to the length of the eccentric rods; that is, equal to the distance from the center of the eccentric to the center of the link block when in full gear, or the distance *F d* in Fig. 466, or *F e'* in Fig. 468. This is the length by which the rods are represented in all link diagrams.

The purpose of curving the link is to make the valve move equally on both sides of a fixed point, no matter what is the position of the link. The radius above does not exactly accomplish this, but it is nearly correct. The effect of curving the link will be clearly seen from Fig. 471. The solid lines show the positions of the link in mid-gear when the crank is on each center. If the link were straight its two positions would be at *a b c* and *e h g*; the center of travel would be at the point *j*. The two positions of the link, if curved, would be at *a d c* and *e f g*, when the center of travel would fall half way between *d* and *f* or at *i*. Now, it will be observed that the point *i* is also the center of travel

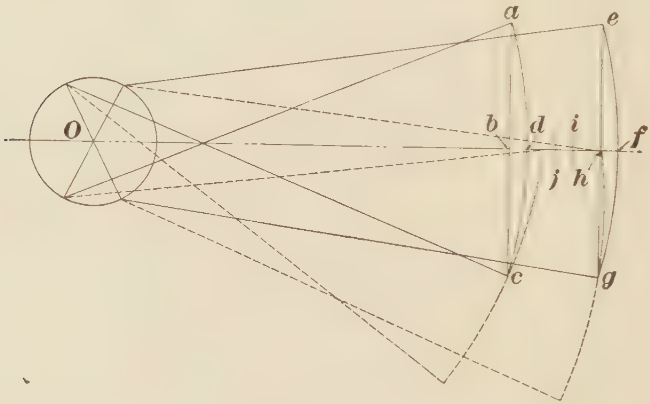


FIG. 471.

for the link when in full gear, as shown by the dotted lines, while the point *j* is to one side of the center. It is necessary, therefore, for the links to be curved.

1662. Eccentric-Rods.—The length of the eccentric-rod is to be taken equal to the distance from the center of the shaft to the middle position of the center of the link arc, or equal to *O i* in Fig. 471. It should not be *exactly* this distance, but any slight error can be corrected in setting the valves.

1663. Laying Out the Motion.—The method of laying out a link motion will be best explained by an

illustrative example. Let the following data be taken: Latest point of cut-off, .9 stroke; full gear lead, $\frac{3}{32}''$; width of steam port, $1\frac{1}{4}''$; width of bridge, $1''$; inside lap, $0''$. Assume the valve to over-travel $\frac{1}{2}''$ in full gear, giving a maximum port opening $1\frac{1}{4} + \frac{1}{2} = 1\frac{3}{4}''$.

Drawing the valve diagram, which is shown in Fig. 472, to a reduced scale, it is found that the travel $= 5''$, outside lap $= \frac{3}{4}''$, and the angle of advance of the eccentric $= 19°$.

From the rule for the width of the exhaust port the latter is found to be $2\frac{1}{2} + 1\frac{1}{4} + 0 - 1 = 2\frac{3}{4}''$. In a locomotive it would ordinarily be made less than this, say $2\frac{1}{2}''$, because in full gear, when the exhaust port is contracted the

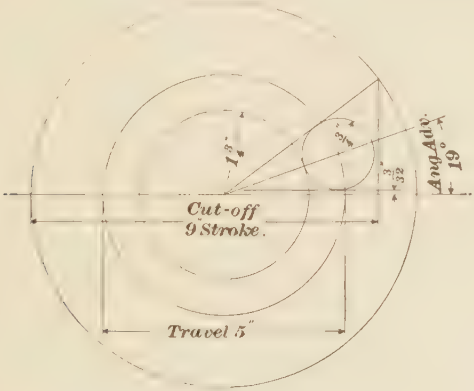


FIG. 472.

greatest amount, the engine is generally drawing a heavy load and runs at a slow speed, thus giving ample time for the steam to exhaust.

1664. In Fig. 473, let O be the center of the shaft and V the relative position of the valve and ports. Draw the center line $a\,x$ of the valve stem, and the center line $O\,C$ of the link motion. In locomotive practice this line usually makes a small angle with the center line of the valve spindle, to allow clearance under the boiler, but in this case we will assume the two to be parallel and that the motion is transmitted from the link to the valve stem through an equal armed rocker, each arm being $9''$ long. The position of the

rocker is usually fixed within certain limits by the design of the engine; in this case the center line will be taken at 4 ft. from the center O of the shaft, or $O\,C = 4$ ft. In the figure, the distance is proportionately shorter than this, so that the whole diagram may be represented on one page.

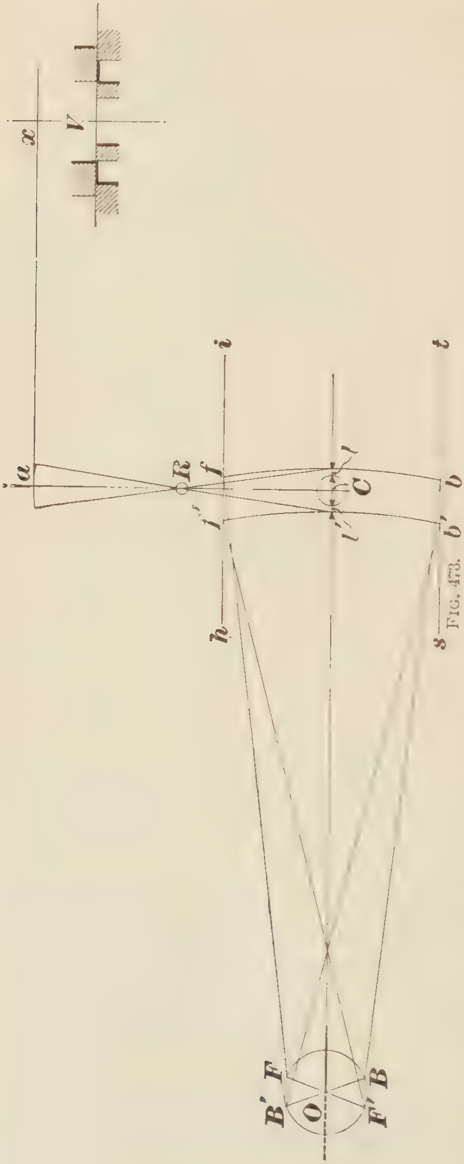


FIG. 473.

To proceed, describe the eccentric circle about O with a radius of $2\frac{1}{2}''$, and draw the center lines of the eccentrics for crank positions on the forward and backward centers. As a rocker is used, the crank leads the eccentric by an angle equal to 90° minus the angle of advance. When the crank is on the forward center, the forward eccentric is at F and the backward at B; when the crank is on the backward center, the forward eccentric is at F' and the backward eccentric at B'.

The length of the link is taken at 13″, and it should now be drawn in full and mid-gear positions, so that its action at these points can be determined. Draw the lines *h i* and *s t* parallel to the center line of motion *O C*, and distant from it $6\frac{1}{2}''$; these will limit the ends of the link when in mid-gear. With *O C* as a radius, and *F* and *B* as centers, describe arcs cutting the horizontal lines just drawn at *f* and *b*; and with the same radius and *F′* and *B′* as centers, describe arcs cutting them at *f′* and *b′*. Now, with the same radius and with centers on the line *O C*, draw the link arcs *f b* and *f′ b′*, which are the positions of the link in mid-gear when the crank is on each center. The mid-gear lead may now be examined by drawing the valve in the two positions corresponding to the above positions of the link, and, if not satisfactory, the proportions of the valve should be changed. An easier and better way, however, is to draw a circle about the point *C*, with a radius equal to the lap of the valve.

Then, since the valve is always displaced from mid-position an amount equal to the lap plus the lead when the crank is on a center, the spaces between the link arcs and the lap circle just drawn, marked *l* and *l′* in the figure, will be equal to the lead for each end of the stroke.

1665. To draw the link in full gear, describe arcs *f* and *f′*, Fig. 474, cutting the center line, with radii

FIG. 474.

equal to *O C* and with centers *F* and *F′*, respectively. With the points of intersection at *f* and *f′* as centers, and radii equal to the length of the link, describe arcs *b* and *b′*, upon

which the lower end of the link must fall. Finally, with B and B' as centers, draw the link arcs $f\,b$ and $f'\,b'$. By drawing the lap circle about C, it may be determined how much the lead varies in full gear. Generally the variation can be compensated for by adjusting the length of the valve stem.

GOVERNORS.

1666. A **governor** is used to regulate the speed of a motor by varying the amount of energy supplied to it. In the water-wheel, for example, it raises or lowers the gate, thus supplying more or less water, and in the steam engine it varies either the quantity or pressure of the steam used. It is driven by the motor it regulates, and is usually so constructed that any variation in the speed of the motor will cause the governor to automatically regulate the speed. In most governors use is made of the centrifugal force of some rapidly revolving body, counteracted by some other force, as gravity or the tension of a spring.

PENDULUM GOVERNORS.

1667. One of the oldest and most common forms of governors is known as the pendulum governor, which is based upon the principle of a revolving pendulum. This form will be taken up first.

1668. A **simple revolving pendulum** is shown in Fig. 475. It may be considered to consist of a ball B suspended by a cord from the point O, the ball revolving about the vertical axis $O\,C$. *Theorctically*, a *simple* revolving pendulum is one having the mass of the ball B concentrated at its center and the cord or arm $O\,B$ of material without weight.

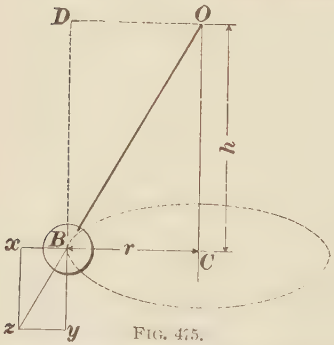


FIG. 475.

When the pendulum revolves about the axis at a uniform speed, the ball remains at a

constant distance r from the axis and at a constant distance $O\,C$ below O, the point of suspension. The latter distance is called the *height* of the pendulum, and is represented by h in the figure. Now, suppose the pendulum to be revolved at a greater speed; the centrifugal force, and, hence, r, will be increased and h will be correspondingly diminished.

When the pendulum is revolving there are three forces acting, namely: Gravity, which is equal to the weight of the ball and always acts downwards; centrifugal force, which acts horizontally outwards, and the tension in the cord. These three forces may conveniently be represented by the parallelogram of forces, as in Fig. 475, where $B\,y$ represents the weight of the ball, $B\,x$ its centrifugal force, and $B\,z$ the tension of the string. Of these, the weight $B\,y$ tends to turn the pendulum about the point O in a vertical plane towards the axis $O\,C$, while the centrifugal force $B\,x$ tends to turn it about O in a direction away from the axis.

Now, in order that the ball shall poise at a certain distance r from the axis, the moment of its weight about O must equal the moment of its centrifugal force about the same center, or, stated in the form of an equation,

$$B\,y \times O\,D\;(= B\,y \times r) = B\,x \times h.$$

But, letting the weight $B\,y$ of the ball $= B$, the centrifugal force $B\,x$ will be $.00034\,B\,r\,N^2$ (formula **19**, Art. **903**), where N is the number of revolutions.

Hence,

$$B \times r = .00034\,B\,r\,N^2 \times h, \text{ or}$$

$$h = \frac{1}{.00034\,N^2} \text{ feet} = \frac{35,294}{N^2} \text{ inches.} \qquad \textbf{(158.)}$$

The height of a revolving pendulum, therefore, is independent of the weight of the ball or the length of the string, and depends solely upon the number of revolutions.

1669. Pendulum Governor.—Figs. 476 and 477 show two forms of the pendulum governor, as used on certain classes of steam engines, and they differ in principle

from the simple revolving pendulum only in that the governor balls are connected with arms, links, etc., the weight of which modifies their action.

1670. In Fig. 476 the balls B, B are suspended from the collar C_1; this has the same effect as though they were suspended from the point O, at the intersection of the center lines of the arms. Two balls are used for the sake of symmetry and even action. The links l, l connect the arms with a lever collar C_2, which is free to turn in an annular groove in a sleeve U which can not turn, but can slide up and down on the spindle S. The sleeve extends into the standard M, and is connected with the rod R_1 by a stud working in a slot

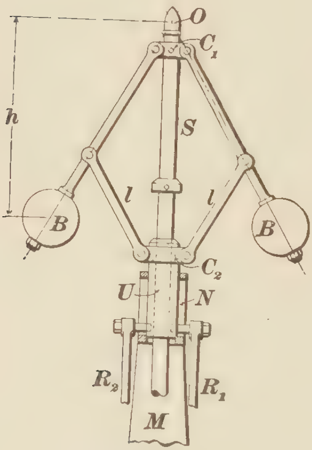


FIG. 476.

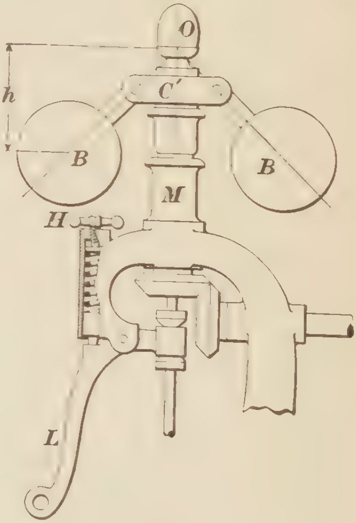


FIG. 477.

N. The spindle is driven by the engine through a belt and a pair of miter gears which are not shown. As the collar C_1 is pinned to the spindle, the balls revolve with the spindle, and, consequently, fly outwards, raising the rod R_1 by means of the links l, l, the collar C_2, and the sleeve M. On Corliss engines, on which this style of governor is largely used, the rod R_1 operates the mechanism that trips the steam valves

at the point of cut-off, thus regulating by varying the time during which steam is admitted to the cylinder.

1671. To prevent sudden fluctuations of the governor, a second rod R_2 operates a piston in a cylinder closed at both ends and filled with oil. The piston consists of two plates through which holes are drilled, and by turning the plates one way or the other they can be adjusted to allow more or less opening through the holes. This adjustment determines the freedom with which the oil can pass from one side of the piston to the other, and, hence, the freedom with which the piston can move. Other constructions are also used for the same purpose.

1672. Fig. 477 shows a throttling pendulum governor used to regulate an engine by varying the pressure of the steam. M is a part of the frame and is bored for a bearing in which turns a hollow spindle driven by miter gears. Inside of the spindle is the valve stem, the lower end of which can be seen. The balls are suspended from the collar C', which receives its motion from the spindle, and the arms to which the balls are attached are in the form of bell-cranks with their upper arms extending inwards and revolving in a groove about the end of the valve. As the speed increases the balls fly out and the upper arms of the bell-cranks lower the valve stem, thus partially closing the valve and shutting off the steam; with a decrease of speed the above operation is reversed. A cord can be attached to the lever L and carried to a place within convenient reach. Then, should the governor belt run off or break and the engine begin to "race," by pulling the cord the valve stem would be lowered and the engine brought to normal speed. The screw H is for regulating the distance the spindle can be lowered; that is, it fixes the lowest speed at which the engine can run. The governor in Fig. 477 is a Gardner governor.

1673. The height of a pendulum governor varies with the speed, and, as in the case of the simple revolving

pendulum, it is the perpendicular distance between the center of the governor balls and their equivalent point of suspension at the center line, or h in Figs. 476 and 477. It is evident that the height at which the balls stand at any particular speed determines the position of the regulating device for that speed, as the rods in Fig. 476 or the valve stem in Fig. 479.

Take, for example, the case of a throttling governor on a steam engine. When the engine is running "light," without any load, the governor valve will be open just wide enough to admit steam to the cylinder at the pressure necessary to keep the engine at the proper speed. When the engine is running "loaded," however, the valve must be opened wider. Now, this variation in the opening of the valve can be caused only by a variation in the height of the governor, which, in turn, is due to a change in speed. Hence, the governor cannot control the speed except within certain limits, which manifestly should not be far apart. A well-designed engine will not vary more than two per cent. either way from a certain mean speed, and the difference in the heights of the governor due to these two extremes of speed must be sufficient to move the throttle valve through its full range of action.

1674. To show clearly what may be expected of an ordinary pendulum governor, Table 36 has been prepared.

TABLE 36.

Revolutions.	Height in Inches.	Variation in Height, Inches.
200	.882	.035
150	1.569	.06
100	3.529	.14
50	14.118	.57

In the second column the approximate heights of a

pendulum governor at different speeds are given, computed by the formula

$$h = \frac{35,294}{N^2}.$$

In the third column the variation in height for a speed variation of 2% each way, or a total variation of 4% or $\frac{1}{25}$ of the mean number of revolutions, is given.

From the table we observe that at high speeds the heights and variations are very small. Thus, for a speed of 200 revolutions, the height is .882 inch, and it would be difficult to construct the governor. At 100 revolutions the height would be 3.529 inches. In this case, also, there would be constructive difficulties, especially if the governor were to be like Fig. 476. The allowable variation for this speed, moreover, is only .14 inch, a very small amount to control the working of a cut-off mechanism, or throttle valve, throughout its whole range of action.

1675. Weighted Pendulum Governor. — The weighted pendulum governor, or Porter governor, is designed to run at an increased height for a given speed, and to have a greater variation for a given variation in speed. This latter is called increasing the *sensitiveness* of the governor; a governor, for example, whose height varies one inch for 2% change in speed is more sensitive to that change than one whose height varies only one-half inch. In the Porter governor, the counterpoise weight is free to revolve and slide upon the spindle with the collar. It, therefore, adds to the weight of the balls by falling down through the links, or arms, but does not add to their centrifugal force, the result being that the height of the governor for any speed is increased.

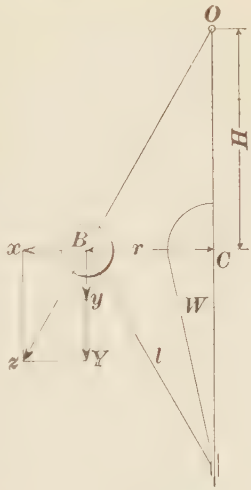


FIG. 478.

1676. In Fig. 478 the weight of the ball B is represented by the line $B\,y$, and its centrifugal force by $B\,x$. Suppose the effect of the counterpoise W is to add an additional weight $y\,Y$ to the weight of the balls. The arm will then have a direction $B\,z$, and the height of the governor will be H. Let $B = y\,B$ and $y\,Y = W$, whence, $B\,Y = B + W$. Then, taking moments about O, as in the case of the simple pendulum governor, we have for equilibrium

$$B\,x \times H = (B + W) \times r;$$

in which $W =$ entire weight of counterpoise and $B =$ weight of one ball.

But $B\,x = .00034\,B\,r\,N^2$ (formula for centrifugal force).

Hence, $(B + W)\,r = .00034\,B\,r\,N^2\,H$, or,

$$H = \frac{1}{.00034} \times \frac{\left(1 + \frac{W}{B}\right)}{N^2} \text{ feet} =$$

$$\frac{35{,}294\left(1 + \frac{W}{B}\right)}{N^2} \text{ inches.} \qquad \textbf{(159.)}$$

By reference to formula **158,** it will be seen that the height of a weighted governor equals the height of a simple governor multiplied by $\left(1 + \frac{W}{B}\right)$.

It follows from this, moreover, that for a given variation in speed, the variation in height will be correspondingly increased, making the governor more sensitive.

From formula **159,** it is evident that adding to the weight of the counterpoise W will increase the height of the governor. That is, the balls will drop lower, and the speed of the motor will increase until the centrifugal force of the balls is sufficient to restore equilibrium again. We thus have a method of increasing or diminishing the speed at which a motor will run by adding to or subtracting from the weight of the counterpoise.

1677. Spring governors are frequently used on throttling engines and other motors to regulate the opening of the valve. They are simply pendulum governors, with springs added to resist the action of the centrifugal force, thus increasing the height and sensitiveness in the same way that the weight operates with the Porter governor. Fig. 479 shows a Pickering governor with springs s, s attached to the spindle l and bearing against the governor arms. This governor has been previously described (see Art. **1290**).

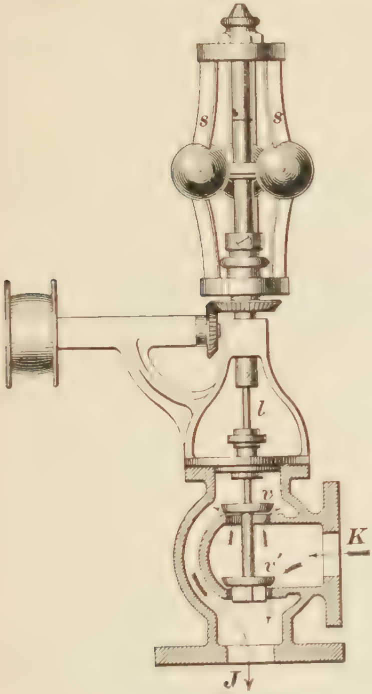


FIG. 479.

TO DESIGN A WEIGHTED GOVERNOR.

1678. Designing a governor consists mainly in so proportioning the parts that the forces acting will balance. It is convenient to assume the weight of the counterpoise as acting at the points B, B' in Fig. 480. W is the counterpoise, the weight of which is represented graphically by $a\ b$. Completing the parallelogram of forces, we have $a\ c$ as the pull upon the lower arm l. Laying this off at $B\ c'$ and again completing the parallelogram, we have $B\ b'$ as the resulting downward pull at B. If the upper and lower arms are *equal* (when continued to the center line) as in the figure, $B\ b'$ will equal $a\ b$; that is, the effect of the counterpoise acting at a will be the same as though its weight were transferred to each ball, making $2\ W$ when acting from B

and B'. If the upper and lower arms are not equal, the value of $B\,b'$ can easily be found by drawing the parallelograms. In what follows the arms will be considered equal.

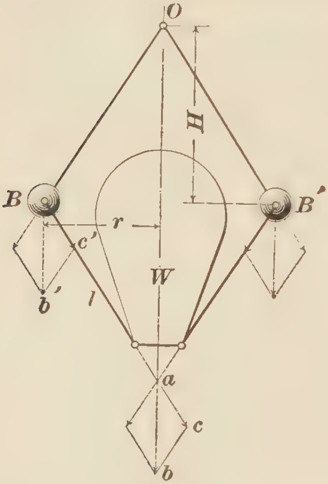


FIG. 480.

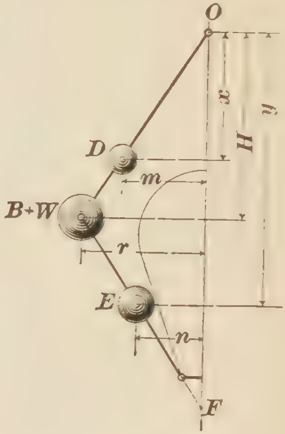


FIG. 481.

1679. In any governor the weight and centrifugal force of the arms exert an influence upon its action that can not be neglected. The effect of the arms is the same, however, as though their weights were concentrated at their centers of gravity, as at D and E in Fig. 481. Here there are four weights to be considered; the weight of the ball, counterpoise, and the two arms, which we will call B, W, D, and E, respectively. For a condition of equilibrium it will be necessary that the sum of the moments of these weights about O shall equal the sum of the moments of the centrifugal forces of B, D, and E, as in the simple case shown in Fig. 478. The sum of the moments of the weights is evidently

$$(B + W)\,r + D\,m + E\,n. \quad (a)$$

For the sum of the moments of the centrifugal forces we have, from the formula for centrifugal force, $.00034\,D\,m\,N^2\,x + .00034\,B\,r\,N^2\,H + .00034\,E\,n\,N^2\,y$. This, reduced, and with the dimensions expressed in inches, becomes

$$.000028\,N^2\,(D\,m\,x + B\,r\,H + E\,n\,y). \quad (b)$$

For equilibrium to exist (a) and (b) must evidently be equal so that

$$(B+W)r+Dm+En=.000028N^2(Dmx+BrH+Eny). \qquad \textbf{(160.)}$$

1680. We have seen that to accomplish regulation the speed of the governor must increase or decrease. Suppose it to increase above mean speed. At first the height will not change, because there will be a certain amount of frictional resistance to overcome. When the speed has reached a certain point, however, the centrifugal force of the balls and arms will have increased enough to change the height, and the governor will regulate.

In ordinary practice it will be safe to assume a variation either way of 2% of the mean speed. Now, let R be the resistance in pounds, which is assumed to act at the collar, N the mean speed, and N_1 the speed just as the resistance is overcome. Then, the governor should be so designed that two per cent. increase of the mean speed will produce an increase in the centrifugal moments in (b) sufficient to balance the moment of the resistance R*, which is $R\,r$.

Hence, from (b),

$$.000028\,[(1.02\,N)^2-N^2]\,(Dmx+BrH+Eny)=Rr, \text{ or}$$

$$.00000114\,N^2\,(Dmx+BrH+Eny)=Rr. \qquad \textbf{(161.)}$$

Finally, by transposing **160** and **161**, we obtain equations for determining the weight of the counterpoise and balls, as follows:

$$W=\frac{1}{r}[.000028\,N^2(Dmx+BrH+Eny)-(Br+Dm+En)]. \qquad \textbf{(162.)}$$

$$B=\frac{R}{.00000114\,H\,N^2}-\frac{Dmx+Eny}{rH}. \qquad \textbf{(163.)}$$

1681. The process is now to first draw the arms, their spread, and the height to be taken, of proportions suited to the size of the engine, or taken from some example at hand.

* The resistance is supposed to be transferred to the center of each ball in the same way that the weight of the counterpoise was treated.

Next, compute the weights and centers of gravity of one upper and one lower arm, as in Fig. 481, taking the weights in pounds, and dimensions in inches. Referring to Fig. 481, suppose they are found to be as follows:

$$x = 5. \qquad r = 8.$$
$$H = 10 \qquad n = 4.$$
$$y = 15. \qquad D = 6 \text{ lb.}$$
$$m = 4. \qquad E = 6 \text{ lb.}$$

Assuming $N = 200$ rev. and $R = 5$ lb., we have from **163**,

$$B = \frac{5}{.00000114 \times 10 \times 200^2} - \frac{(6 \times 4 \times 5) + (6 \times 4 \times 15)}{8 \times 10} =$$

4.96 lb.

That is, each ball must weigh 5 lb., nearly, to cause the governor to operate when the speed has varied two per cent.

From **162**, we have

$W = \frac{1}{8}. \quad [.000028 \times 200^2 (6 \times 4 \times 5 + 5 \times 8 \times 10 + 6 \times 4 \times 15) - (5 \times 8 + 6 \times 4 + 6 \times 4)] = 112.2$ lb.

W should include, beside the counterpoise, all the dead weight coming on the collar. Draw the counterpoise, and find whether it will allow the desired play of the arms. If not, the process must be repeated with other dimensions.

1682. When the governor is built it should be tested, owing to variations that will exist in the weights and sizes and the uncertainty of the resistance R. The counterpoise should be a little too large, so that metal can be turned off for adjustment, and the balls should be cast hollow and filled with lead to facilitate adding or removing weight.

SHAFT GOVERNORS.

1683. Modern high-speed engines, in which slide valves of one form or another are almost invariably employed, are regulated by governors that act upon the eccentrics and vary the points of cut-off according to methods already described. The governor is placed on the main shaft of the engine, from which it derives its name. It consists of

revolving weights whose centrifugal force is entirely balanced by springs.

Three different shaft governors will be described, each of which shifts its eccentric in a different way. No two makers of high-speed engines use shaft governors exactly alike; but if the principles of a few of them are understood, the student should have no difficulty with any.

1684. Buckeye Engine Governor.—The Buckeye engine has a special form of hollow slide valve, with cut-off valve inside, and is regulated by a governor that varies the point of cut-off by changing the angle of advance. The arrangement for effecting this is shown in Fig. 482. A governor wheel for supporting the parts of the governor is

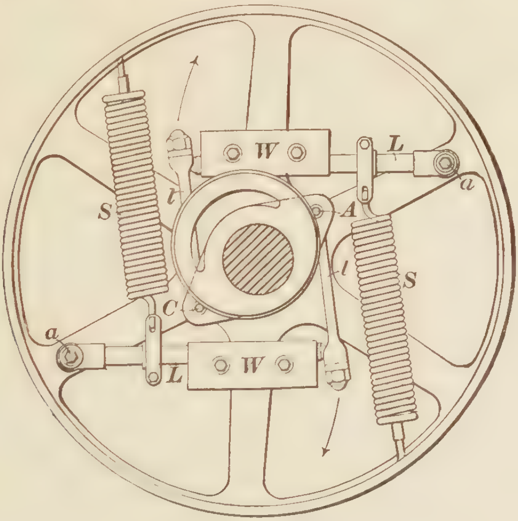


FIG. 482.

keyed to the shaft. *L*, *L* are two arms pivoted to two arms of the wheels at *a* and *a*. Two links *l*, *l* join these arms with lugs on the collar *A C*. The cut-off eccentric is fast to this collar and is loose on the shaft, while the eccentric for the main valve is keyed to the shaft. Now, as the speed increases the centrifugal force of the weights *W*, *W* increases, causing the ends of the arms *L*, *L* to fly out

towards the rim of the wheel and the eccentric to turn on the shaft. The centrifugal force of the weights is balanced by the tension of the springs *S*, *S*.

With a single-valve engine this form of governor could not be used, because, as we have seen, the variation of the angle of advance would produce too great a variation of the lead. But in the Buckeye engine, the admission of the steam is not governed by the cut-off valve, so that in this case, this form of governor is as good as any.

1685. Erie Engine Governor. — This governor, shown in Fig. 483, regulates the cut-off by moving the eccentric across the shaft. A frame *A C* is keyed to the shaft, and two bell-crank levers *L*, *L*, pivoted to the frame, carry the weights *W*, *W* at one end and at the other end are connected with the eccentric in the manner shown. As the weights fly out, the eccentric is moved down (in the figure), the angle of advance is increased, the throw of the eccentric diminished and the cut-off shortened. The spring *S*, by being compressed, resists the action of the weights.

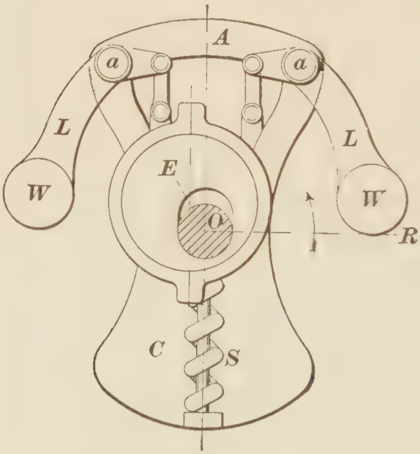


FIG. 483.

It will be well to notice here that in this governor the eccentric moves in a line at right angles to the center line *O R* of the crank. Why this is so will be clear upon reference to the diagram for this form of shifting eccentric in Fig. 461. There the eccentric is supposed to move in the line *D E* parallel to *A C*. But *A C* is the line from which the angle of advance is laid off. Hence, as when no rocker is used, the angle between the crank and eccentric must be

$90° +$ the angle of advance; the relative position of the crank must be at right angles to $A\ C$ and below it, or at right angles to line $D\ E$ in which the eccentric moves. As the Erie engine has no rocker, the relative crank and eccentric positions must be $O\ R$ and $O\ E$, and the eccentric moves at right angles to the crank.

1686. Straight-Line Engine Governor.—Fig. 484

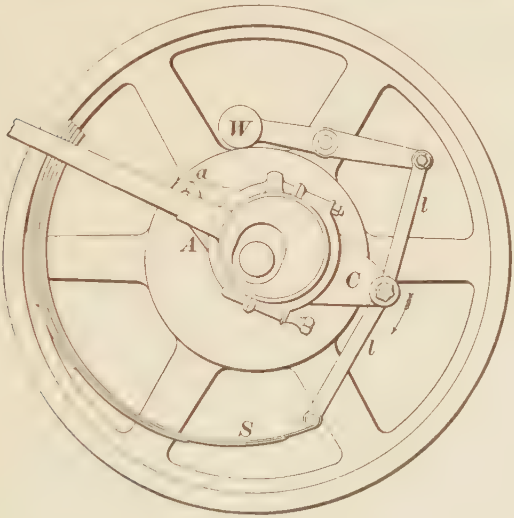


FIG. 484.

shows the principle of the governor used on the Straight-Line engine. The eccentric is on a plate $A\ C$, pivoted at a. As the weight W flies out, the eccentric is shifted about the center a, the links l, l moving in the direction of the arrow and compressing the flat spring S. In this case the governor is attached directly to one of the fly-wheels of the engine.

STEAM BOILERS.

TYPES OF STEAM BOILERS.

1687. A **steam boiler** is a closed vessel in which steam is generated for power or heating purposes.

The boiler when in use is but partially filled with water, thus dividing the space within it into two parts.

1688. The **water-line** is the level of the surface of the water in the boiler.

1689. The **steam space** is the space in the boiler above the water-line.

1690. The **heating surface** of a boiler is the part of its surface that is exposed to the fire and to the hot gases from the fire as they pass from the furnace to the chimney.

1691. The **fittings** of a boiler consist of such attachments as gauges for showing the steam pressure and amount of water in the boiler, safety valve, steam stop-valve, etc.

1692. In accordance with the different conditions under which boilers are used, they may be roughly divided into three classes—**stationary, locomotive,** and **marine boilers.** Each class is made in a great variety of forms that depend on the surroundings, first cost, quality of fuel and water used, and the steam pressure required.

1693. The **plain cylindrical boiler** is shown in Figs. 485, 486, and 487. It consists essentially of a long cylinder called the **shell,** which is made of iron or steel plates riveted together as shown in Fig. 485; and the ends of the cylinder are closed by flat or hemispherical plates called the **heads** of the boiler. One of the heads is shown in Fig. 486, carrying the fittings B, C, C_1, and C_2. In this type of boiler the heads are often made of thick cast iron,

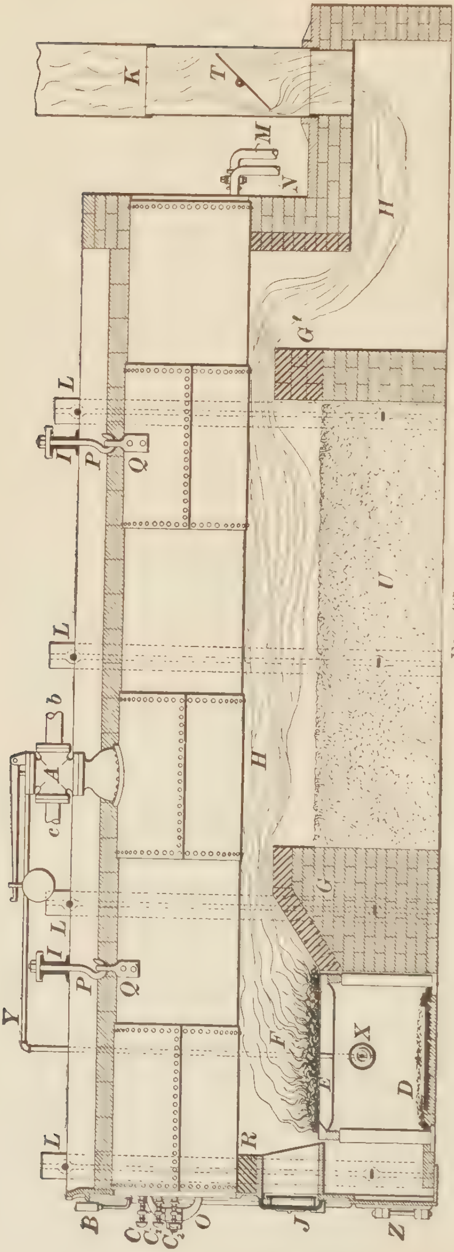


FIG. 485.

though wrought-iron plate may be employed; the hemispherical or dished form of head is generally used, since it is stronger than the flat head.

The manner of suspending the boiler is clearly shown in Figs. 485 and 487. The boiler is enclosed by side walls of brick. The channel beams *I*, *I* are laid across these brick side walls, and the boiler is suspended from these beams by means of the hooks *P*, *P* and eyes *Q*, *Q* (see Figs. 485 and 487), the latter being riveted to the shell.

The side walls are supported and prevented from buckling by the **binders** or **buckstaves** *L*, *L* bolted together at the top and the bottom. The buckstaves are cast-iron bars of **T** section, as shown in the figure. The

eyes Q, Q are placed about one-fourth of the length of the shell from each end. This method of suspending the shell allows it to expand and contract freely when heated or cooled.

The rear wall is built around the rear end of the shell, as is shown in Fig. 485, and continued back to form the chamber H, into which opens the chimney or stack K. The **boiler front,** shown in Fig. 486, is of cast iron. Fig. 485 shows a section of the front. The front end of the shell is partly surrounded by the firebrick R. The weight of the

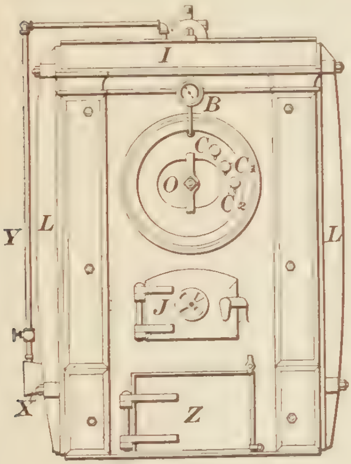


FIG. 486.

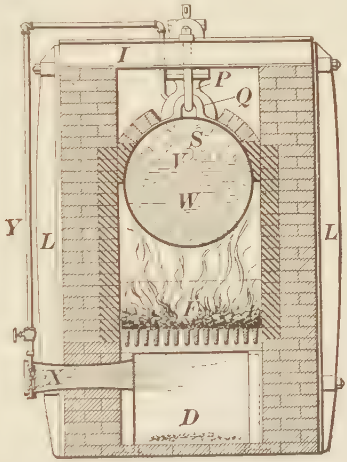


FIG. 487.

shell comes upon the hooks P, P, the rear wall and firebrick R simply keeping it in position.

1694. The furnace F is placed under the front end of the boiler shell. The fuel is thrown in through the furnace door J and burns upon the grate E, the ashes falling through the grate into the ash pit D. To insure a supply of air sufficient for the complete combustion of the fuel, the furnace is sometimes supplied with the **blower** X. It consists of a cylinder leading into the ash pit D, into which is led a jet of steam through the pipe Y. The jet rushes into the ash pit with great velocity, and carries a quantity of air with it, in much the same way as the locomotive blast, described in

Art. **1077,** carries the air from the space in front of the tubes out through the stack. The pressure of the air in the ash pit is thus increased, more air is forced through the fire, and the combustion of the fuel is more rapid and complete.

1695. Behind the furnace is built the brick wall G called the **bridge,** which serves to keep the hot gases in close contact with the under side of the boiler shell. As boilers of this type are generally quite long, a second bridge G' is usually added. The gases arising from the combustion of the fuel flow over the bridges G and G' into the chamber H, and escape through the chimney K. The flow of the gases is regulated by the damper T placed within the chimney. The space U between the bridges is filled with ashes or some other good non-conductor of heat. The door Z in the boiler front gives access to the ash pit for the removal of the ashes. The tops of the bridges, the inner surface of the side walls and rear wall, and, in general, all portions of the brickwork exposed to the direct action of the hot gases are made of firebrick (shown in Figs. 485 and 487 by the dark section lining), since it is able to withstand a very high temperature.

It will be seen by referring to Fig. 487 that the firebrick work covers the upper portion of the boiler shell in such a manner as to prevent the hot gases from coming in contact with the shell above the water-line V. It is a general rule in boiler construction and setting that *under no circumstances should the fire line be carried above the water-line.*

The top of the shell is covered by brickwork or some other non-conducting material to prevent radiation of heat. Water is forced into the boiler through the feed-pipe N which leads from a pump or injector. When in operation the water stands at about the level V, the space S above being occupied by the steam. The *safety valve* is shown at A. The office of the safety valve is to prevent the steam pressure from rising above the desired point. The pipe b is the main steam pipe leading to the engine; the pipe c

provides for the escape of the waste steam when the safety valve blows off. The steam gauge B indicates the pressure of the steam in the boiler. This gauge is attached to a pipe which passes through the front head into the steam space.

The **gauge-cocks** C, C_1, and C_2, placed in the front head of the shell, are used to determine the water-level. For instance, if the cock C_1 is opened and water escapes, it is evident that the water-line is above the cock C_1, while, if steam escapes, the level must be below C_1

The **manhole** O is a hole in the front head through which a man may enter and inspect or clean the boiler. The hole is closed by a plate and yoke.

To permit the boiler to be emptied, it is provided with a **blow-off pipe** M, through which the water or sediment may be discharged.

The safety valve A, the steam gauge B, the gauge-cocks C, C_1, C_2, the blow-off M, and the manhole O are fittings which will be considered in detail later.

Plain cylindrical boilers are usually made from 30 to 42 inches in diameter, and from 20 to 40 feet long, though in rare instances they have been constructed with a diameter of 48 or more inches, and a length of 60, or even 100, feet.

They are much used in mining districts where fuel is very cheap; but, on account of their small heating surface, they are very uneconomical, and, consequently, are not generally used where fuel is expensive. The advantages of this type of boiler are: Cheapness of construction, strength, durability, and ease of access for cleaning and repairs.

1696. The **flue boiler** differs from the plain cylindrical boiler in having one or more large flues running lengthwise through the shell, below the water-line.

Such a boiler is shown in elevation and section in Figs. 488, 489, and 490.

The ends of the flues A, A are fixed in the front and rear heads of the shell, respectively. The front end of the shell is prolonged beyond the head, forming the **smokebox** B, into

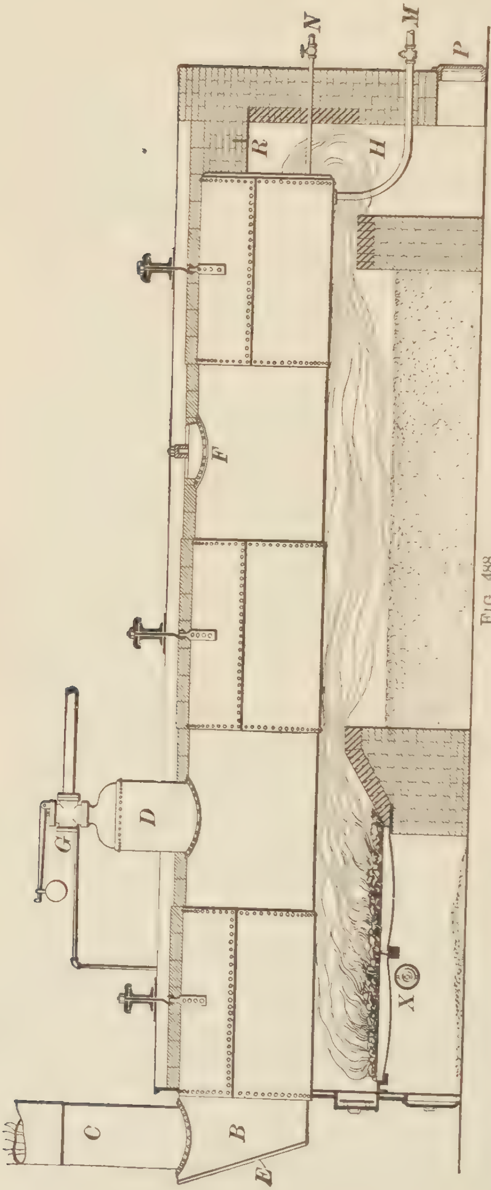


FIG. 488.

which opens the smokestack *C*. The front of the smokebox is provided with a door *E*. The boiler shell is also provided with the **dome** *D*, which forms a chamber where steam may collect and free itself from its entrained water before passing to the engine. The manner of supporting the shell, and the construction of the furnace and bridges, are the same as was described for the plain cylindrical type. The hot gases, however, pass over the bridges to the chamber *H*, and then back through the flues *A*, *A* into the smokebox *B*, and out of the stack *C*. It is plain, therefore, that the heating surface is greater than that of the plain cylindrical boiler by the cylindrical surface of the flues *A*, *A*.

As shown in Fig. 489, the boiler has a cast-iron front, to which the furnace door and ash-pit doors are attached.

The safety valve *G* is attached to the top of the dome; from the safety valve are led the two steam pipes, one to the engine, the other to carry the escaping steam outside the building.

The steam gauge *K* and gauge-cocks are placed on a column *L* which communicates with the interior of the shell through the pipes *s* and *t*, the former entering the steam space and the latter the water. The manhole *F* is placed on

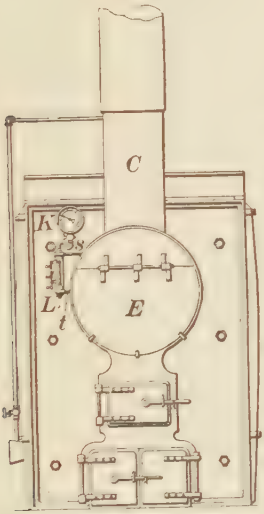


FIG. 489.

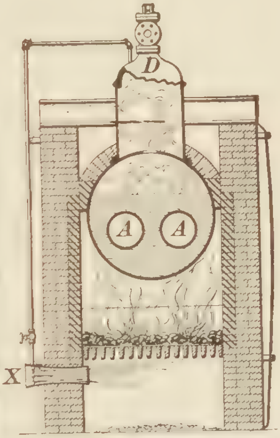


FIG. 490.

top of the shell instead of in the head. The feed-pipe is shown at *N*, the blow-off pipe at *M*; both pass through the rear wall. Access is given to the rear end of the shell and to the pipes *M* and *N* through the door *P*.

This form of boiler may be provided with a blower as shown at *X*, Figs. 488 and 490.

The brick wall is built and supported in about the same manner as the wall of Fig. 485. The cast-iron **flue plate** *R* rests on the side and rear walls and supports the brickwork above it.

1697. The Return Tubular Boiler.—This type of boiler is a development of the flue boiler, the two large flues of the latter being replaced by a large number of small tubes. The object of introducing the numerous tubes is to increase the heating surface of the boiler.

A side view of a tubular boiler is shown in Fig. 491; a

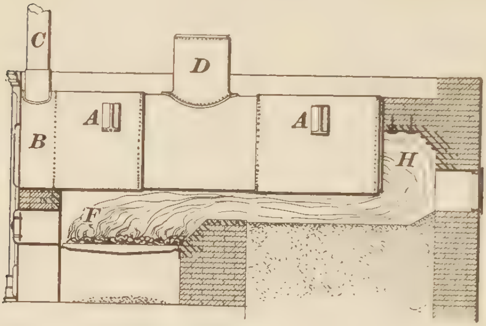


FIG. 491.

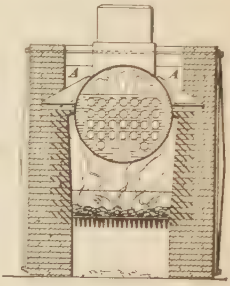


FIG. 492.

cross-section through the tubes is shown in Fig. 492. The tubes extend through the whole length of the shell, the ends being *expanded* into holes in the heads of the boiler. The front end of the shell projects beyond the head forming the smokebox *B*, into which opens the stack *C*.

The shell is supported by the side walls through the brackets *A*, *A*, which are riveted to the shell. The boiler is usually provided with a dome *D*, though this is sometimes left off. The walls are built and supported by buckstaves in practically the same manner as those previously described. Since this type of boiler is generally short, one bridge only is used. Firebrick is used for all parts of the wall exposed to the fire or heated gases. The fittings are not shown in the figure. The safety valve would be placed on top of the dome, and the pressure gauge and gauge-cocks would be placed on the front. The manhole is either in one of the heads or on top of the shell. The feed-pipe may enter the front head, the rear head, or the bottom of the rear end of the shell, while the blow-off pipe is placed at the bottom of the shell at the rear end. Access is given to the rear end of the boiler through the door *E*.

As usual, the furnace *F* is placed under the front end of the boiler. The gases pass over the bridge, along under the boiler into the chamber *H*, then back through the tubes to the smokebox *B*, and out of the stack *C*.

The return tubular boiler is probably more used in the United States than any other. The details of its construction and setting will be shown later.

1698. Cornish and Lancashire Boilers.—In the three forms of boilers so far considered, the furnace is placed outside of the shell of the boiler; such boilers are said to be **externally fired.** Upon the invention of the single flue boiler, the idea was conceived of placing the fire in the flue, and the result is the so-called **Cornish boiler,** a cross-section of which is shown in Fig. 493.

The boiler is set in masonry in such a manner as to form the passages *A*, *A*, and *B*. The grate is supported in the single large flue *C*. The heated gases pass from the furnace to the rear through the flue *C*; they then return beneath the boiler through the flue *B*, and finally return to the rear through the side flues *A*, *A*, and thence out of the chimney. This path of the gases constitutes the **split draft.**

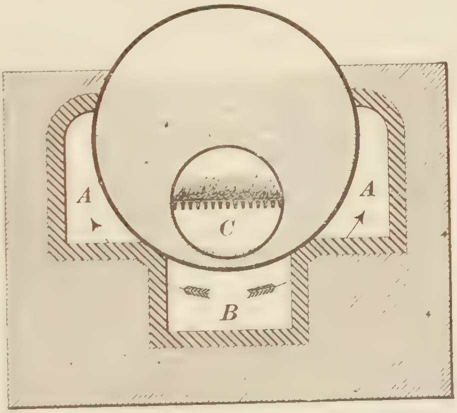


FIG. 493.

It was formerly the general practice to arrange the brickwork setting so that the gases returned to the front through the side flues *A*, *A*, and returned to the rear through the lower flue *B*. It was found, however, that this practice retarded the circulation of the water and rendered the shell

more liable to strains due to unequal expansion and contraction. Consequently, the first method of producing the split draft is used almost exclusively in modern practice.

As shown in the figure, the brickwork passages are lined with firebrick.

1699. The **Lancashire boiler** is a modification of the Cornish type. In order to give a large grate area and a large heating surface for the same diameter of shell, two large furnace flues are substituted for the one flue of the Cornish type. The brickwork setting (Fig. 494) is precisely similar to that of the Cornish boiler, Fig. 493; the split draft is also formed in the same manner.

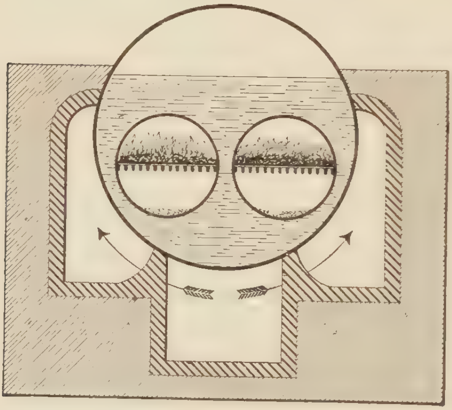


FIG. 494.

1700. The large furnace flues of internally fired boilers, of which the Cornish and Lancashire are examples, are subjected to an external collapsing pressure, equal to the pressure of the steam. The greater the diameter of the flue, the more liable it is to collapse; consequently, the Lancashire possesses an advantage over the Cornish type in this respect, since each of its two flues is necessarily of smaller diameter than the single flue of the Cornish boiler. Various measures are taken to strengthen the furnace flues of internally fired boilers. They are sometimes made corrugated, as shown in Fig. 496; again, they have channel irons riveted around them. A very common method, however, is to stiffen them by transverse conical water legs, as shown in Fig. 495. The water legs not only stiffen the flue, but also provide an opportunity for the circulation of the water and split up the gases on their way

through the flue, thereby providing an increase of heating surface.

1701. The **Galloway boiler** is a sort of combination of the Cornish and Lancashire types. It has two internal furnace flues fitted with grates, ash pit, etc., in the usual manner. Instead of extending through the whole length of shell, the two flues unite just behind the bridge into one large kidney-shaped flue which extends from this junction to the rear head of the shell. This large flue is strengthened by a large number of water legs of the form shown in Fig. 495. The setting of the Galloway boiler is similar to that shown in Figs. 493 and 494. The draft is split as previously described.

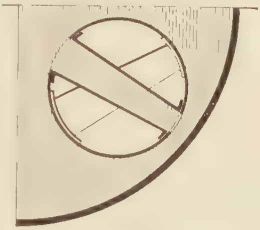


FIG. 495.

1702. The Cornish, Lancashire, and Galloway boilers belong to the general class known as **internally fired** boilers. The chief objection to these boilers is the liability of the collapse of the internal flue, and the straining actions set up by the expansion and contraction of the flue. The chief point in favor of these boilers, and in favor of internally fired boilers generally, is their economy in the use of fuel. Generally speaking, all conditions being the same, an internally fired boiler is 10 per cent. more economical than an externally fired boiler. This fact is due to the loss of heat by radiation through the brickwork setting of the latter class of boilers.

The three types of boilers just described are extremely popular in England, and on the continent of Europe, but they are little used in this country.

1703. A stationary boiler combining the features of the Lancashire and multitubular types is shown in Fig. 496. It consists of a large cylindrical shell which is short in comparison with its diameter. Two large furnace flues *A*, *A* (only one of which is shown in the figure) extend side by side lengthwise through the shell, the ends of the flues being riveted to the heads of the shell. Above and parallel with

these flues, and below the water-line, a series of tubes *B* extend through the shell from end to end. The front ends of these tubes open into a smokebox *F* which is connected with the chimney or stack.

The boiler is enclosed in brickwork, which, however, does not form the support, but is built to form the chamber *E*

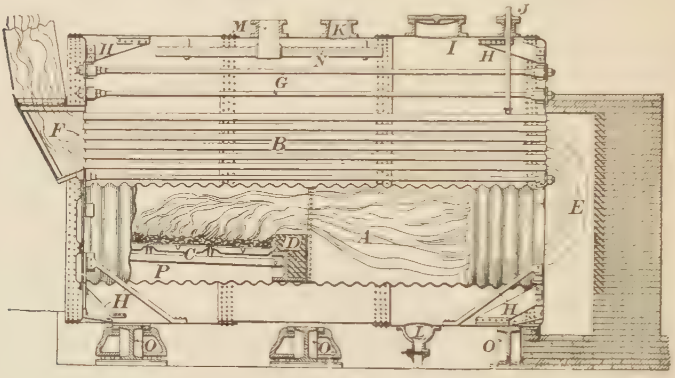


FIG. 496.

and to prevent radiation. The shell is supported on the beams *O*, *O*, *O*.

The furnace flues are corrugated to render them stronger against a collapsing pressure. The flat heads are restrained from bulging and spreading apart by the heavy stayrods *G* and the diagonal braces *H*, *H*.

The furnaces are placed within the large flues, and, as usual, consist of the grate *C*, ash pit *P*, and bridge *D*. The gases arising from the combustion flow to the rear through the flues *A* into the chamber *E* and back to the front through the tubes *B*; thence into the smokebox *F* and out of the chimney. It is evident that this type of boiler has a very large amount of heating surface in proportion to its grate area.

Water is fed into the boiler through the feed-pipe *J*. The various fittings are not shown in the figure. The pressure gauge and gauge-cocks are usually placed on the boiler front; the safety valve is bolted on to the nozzle *K*; the blow-off is at *L*. The main steam pipe is bolted on at *M*.

The steam is collected by the **dry pipe** *N* which is perforated with numerous small holes. The dry pipe is effective in freeing the steam from any unevaporated water which may be mixed with it. It is often used in place of a dome on this form of boiler and also upon the other types of internally fired boilers just described. The manhole is shown at *I*.

1704. The Locomotive or Fire-Box Boiler.—Next to the multitubular type, the fire-box boiler is probably more used than any other type. It is used exclusively in railway service, and also largely as a stationary boiler. A large proportion of the small portable combined engines and boilers used for agricultural purposes have the fire-box type of boiler. The general construction of this type of boiler is shown in Fig. 497. The shell is composed of two differently

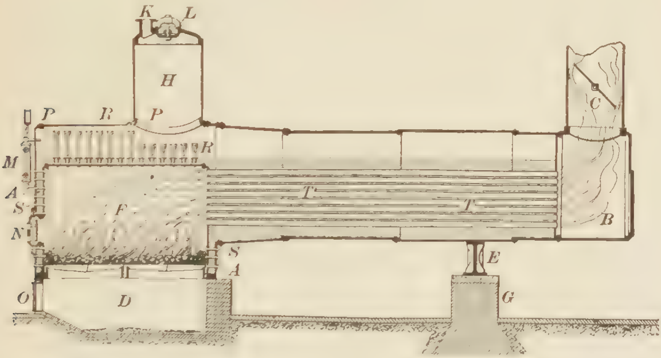


FIG. 497.

shaped parts riveted together. The front part of the shell is cylindrical; the rear part is usually of a rectangular cross-section with vertical sides, or of a trapezoidal section with inclined sides; in either case the top is semi-cylindrical. The furnace *F* is a box of the same shape as the rear end of the shell in which it is placed. A space, called a **water leg**, is left between the sides and ends of the furnace and the shell; this space is filled with water as shown at *A*, *A*. A series of tubes extend from the front sheet of the furnace or **fire-box** to the front head of the shell. The shell is pro-

longed beyond the front head, forming a smokebox *B* into which opens the stack *C*.

As shown in this figure, the spaces *A*, *A*, called *water legs*, only extend down as far as the grate, the ash pit *D* being formed in the brick setting. In many boilers of this type the water legs extend down to the bottom of the ash pit, and sometimes there is a water space below the ash pit; that is, the furnace and ash pit are entirely surrounded by water.

The boiler is supported at the front end by the cast-iron cradle *E* resting upon the masonry foundation *G*. The rear end is supported upon a brick wall which also forms the ash pit. The boiler is usually provided with a dome *H* from which is led the main steam pipe which is bolted on at *K*. In the figure, the dome is provided with a manhole *L*. The feed-water may be introduced at any convenient point in the shell. The pressure gauge, water glass, and gauge-cocks are attached to the column *M* which is placed in communication with the interior of the shell. The furnace and ash-pit doors are shown at *N* and *O*, respectively. The safety valve is usually attached to the dome.

Since the flat sides of the furnace and shell are liable to bulge on account of the pressure, they must be braced or stayed. This is accomplished by the staybolts *S*, *S*.

The flat top of the fire-box is strengthened by a series of parallel girders *P*, *P*. As an additional security, the girders are sometimes attached to the shell by the "sling stays" *R*, *R*.

The gases of combustion pass directly from the furnace through the tubes *T*, *T*, to the smokebox *B*, and out of the stack *C*. In railway locomotives a strong draft is obtained by allowing the exhaust steam to discharge through the smokestack, as described in Art. **1077.**

The tubes of the locomotive boiler are about 12 feet long, 2 inches in diameter, and are made of iron or steel. The tubes of stationary and portable boilers of this type are generally of larger diameter, as there is less demand for great quantities of steam.

1705. The locomotive type of boiler is **self-contained,** that is, it requires no brickwork for flues or for setting.

1706. The Vertical Boiler.—This type is essentially a modification of the locomotive type placed on end. A common form of vertical boiler is shown in Fig. 498. It consists of a vertical cylindrical shell, in the lower end of which is placed a fire-box *F*. The lower rim of the fire-box and the lower end of the shell are separated by a wrought-iron ring *k*, to which both are riveted, the rivets going through both plates and ring. The shell and fire-box are also stayed together by the stay-bolts *a*, *a*. The space between the two is filled with water, so that the fire-box is nearly surrounded by it. The boiler shell, and likewise the grate *E*, rest upon a cast-iron base *D* which forms the ash pit. A series of vertical tubes *t*, *t* extend from the top sheet of the fire-box to the upper head of the shell. The tubes serve as stayrods and strengthen the flat surfaces which they connect. The upper ends of the tubes open directly into the chimney or smokestack *K*. The gases from the furnace thus pass directly through the tubes and out of the stack.

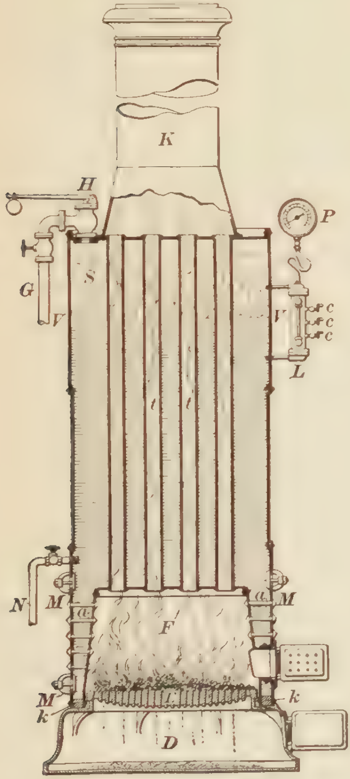


FIG. 498.

The safety valve is shown at *H* with the main steam pipe *G* leading from it. The pressure gauge *P* and gauge-cocks *c*, *c*, *c* are attached to a column *L* which communicates in the usual manner with the interior of the shell. The

construction of this type of boiler does not generally permit the use of manholes, but handholes *M*, *M* are placed in convenient positions for cleaning out mud and sediment.

The boiler is fed through the feed-pipe *N* which is connected to a pump or injector.

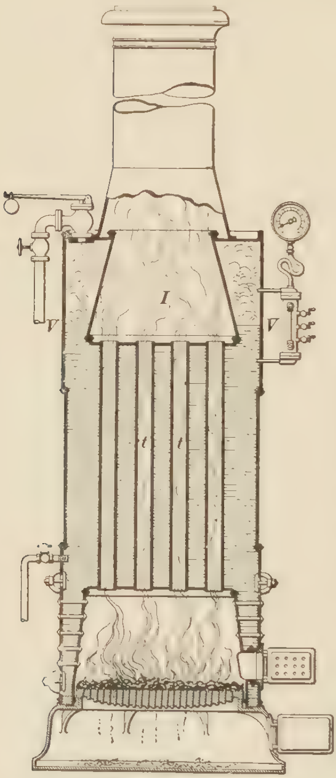


FIG. 499.

When the tubes extend through to the upper head of the boiler, as shown in Fig. 498, their upper ends pass through the steam space *S* above the water line *V V*. This construction has the disadvantage that the upper ends of the tubes are liable to become overheated and to collapse when the boiler is subject to rapid firing.

In the form of vertical boiler shown in Fig. 499, this danger is avoided. A chamber, or smokebox *I*, extends down from the upper head of the shell so that its bottom plate is always below the water-line. The upper ends of the tubes *t*, *t* are expanded into the lower plate of this chamber, thus keeping them surrounded by water from end to end. A vertical boiler constructed in this manner is said to have a *submerged head.* Aside from the submerged head the construction of the boiler of Fig. 499 is similar to that of Fig. 498.

Vertical boilers are generally wasteful of fuel; they are, however, self-contained, require but little floor space, and are easy to construct and repair. For these reasons the vertical type of boiler is very popular with a large class of steam users.

WATER-TUBE AND SECTIONAL BOILERS.

1707. The various types of boilers previously described have been developed from the original plain cylindrical boiler by the addition of flues and tubes, for the purpose of increasing the water-heating surface. A similar development along a different line has given rise to a distinct class of boilers, known as **water-tube boilers.**

1708. The first step in the development was the **double-deck** cylindrical boiler, which consisted simply of two cylindrical shells, the larger placed vertically above the smaller, the two being united by short water legs. The lower, smaller cylinder was placed nearest the furnace, while the upper one contained the steam space.

1709. The **French or elephant boiler** of which Fig. 500 is a cross-section, consists of one large cylindrical shell *M* united to two or more smaller cylinders or heaters *N*, *N* by the water legs *L*, *L*. The heaters *N*, *N* are placed in contact with the fire. The hot gases pass to the rear beneath the heaters and then to the front through the passages *D*, *D*. Consequently, the larger cylinder *M* is exposed only to the comparatively cool gases, while the intense heat of the fire comes in contact with the smaller heaters only, which are much less injured by the alternate expansion and contraction due to the alternate heating and cooling of the furnace. Again, if one of the small heaters is injured it may be more easily replaced.

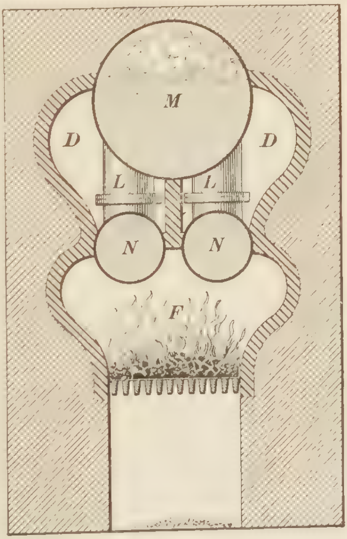


FIG. 500.

There is another important reason in favor of dividing the bulk of water among many small heaters instead of containing it in one large vessel. It may be shown that, with a given pressure of steam per square inch, the bursting pressure tending to rupture the boiler shell is directly proportional to the diameter of the shell.

Suppose, for example, that steam at a pressure of 80 lb. per sq. in. is contained in each of two cylinders, the lengths of which are 10 feet, the diameter of one being 12 in., and of the other 24 in. The total force tending to rupture the smaller shell along a longitudinal seam is $\frac{12 \times (10 \times 12) \times 80}{2}$ $= 57,600$ lb., while the force tending to rupture the large shell in a similar manner is $\frac{24 \times (10 \times 12) \times 80}{2} = 115,200$ lb. That is, the latter is double the former, and, consequently, if the shells were made of the same thickness, the smaller would be twice as strong as the larger.

It follows, therefore, that thin cylinders of small diameter will withstand a high steam pressure as well as thicker shells of large diameter. In general, by using numerous small water cylinders, the boiler may be made lighter for the same capacity, and is less liable to disastrous explosion.

By increasing the number of small heating cylinders, and by substituting for them smaller lap-welded tubes, the modern water-tube boiler has been evolved.

1710. The **Babcock and Wilcox water-tube boiler** is shown in Fig. 501. It consists essentially of a main horizontal drum B and of a series of inclined tubes T, T. (Only a single vertical row of tubes is shown by the figure, but it will be understood that each nest of tubes is composed of several vertical rows.) There are usually 7 or 8 of these vertical rows to each horizontal drum. The front ends of the tubes of a vertical row are all expanded into a hollow iron casting H, called a **header.** The rear ends are expanded into a similar header, and the front and rear headers are placed in communication with the drum by

tubes, or *risers* C and C, respectively. In front of each tube, a handhole is placed in the header for the purpose of cleaning, inspecting, or removing the tubes.

The method of supporting the boiler is not shown in the figure. The usual method is to hang the boiler from wrought-iron girders resting on vertical iron columns. The brickwork setting is not depended upon as a means of support. This make of boiler, in common with all others of the water-tube type, requires a brickwork setting to confine the furnace gases to their proper field.

The furnace is of the usual form, and is placed under the front end of the nest of tubes. The bridge wall G is built up

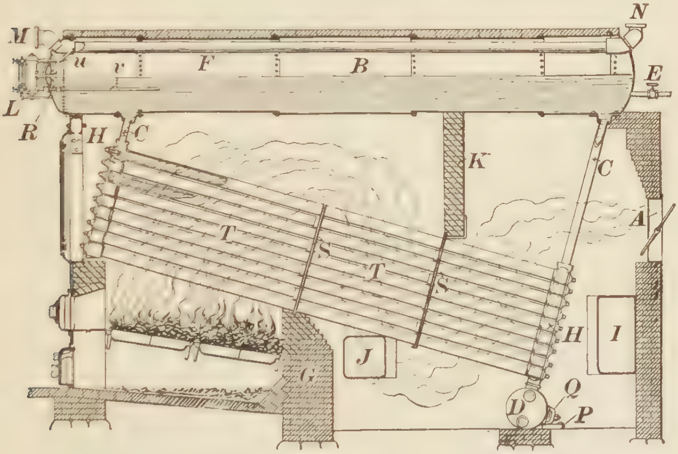


FIG. 501.

to the bottom row of tubes; another firebrick wall K is built between the top row of tubes and the drum. These walls and the baffle plates S, S force the hot furnace gases to follow a zigzag path back and forth between the tubes. The gases finally pass through the opening A in the rear of the wall, into the chimney flue.

The feed-water is introduced through the feed-pipe E. The steam is collected in the dry pipe F which terminates in the nozzles M and N, to one of which is attached the main steam pipe, and to the other the safety valve.

The pressure gauge, cocks, etc., are attached to the column, which communicates with the interior of the shell by the small pipes *u* and *v*, the former of which extends into the dry pipe, the latter into the water.

At the bottom of the rear row of headers is placed the mud drum *D*. Since this drum is the lowest point of the water space, most of the sediment naturally collects there. This sediment may be blown out from time to time through the blow-off pipe *P*. The drum *D* is provided with a hand-hole *Q*, and a manhole *R* is placed in the front head of the drum *B*. The heads of the drums are of hemispherical form,

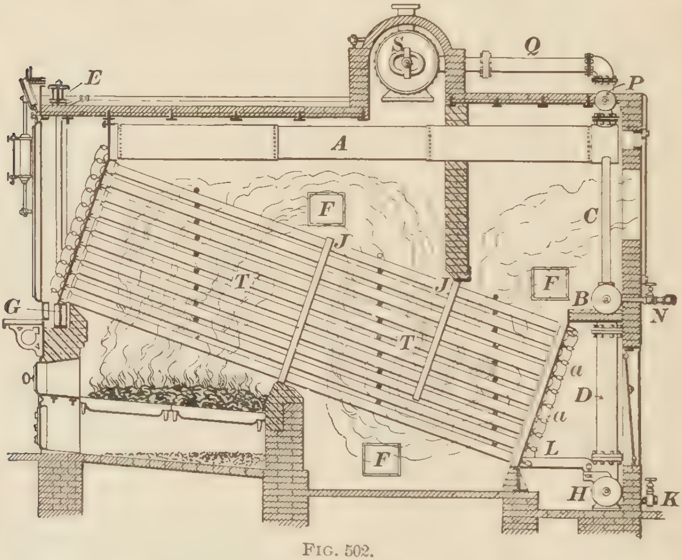


FIG. 502.

and, therefore, do not require bracing. Access may be had to the space within the walls through the doors *I* and *J*.

The circulation of water takes place as follows: The cold water is introduced into the rear of the boiler; the furnace being under the higher end of the tubes, the water in that end expands upon being heated, and is also partially changed to steam; hence, a column of mingled water and steam rises through the front headers to the front end of the drum *B*, where the steam escapes from the surface of the water. In

the meantime, the cold water fed into the rear of the drum descends to the rear headers through the long tubes C to take the place of the water which has risen in front. Thus, there is a continuous circulation in one direction, sweeping the steam to the surface as fast as it is formed, and supplying its place with cold water. Most of the sediment sinks to the mud drum D, from which it is blown out from time to time.

1711. The **Root water-tube boiler** is shown in Figs. 502 and 503, the latter being an end view with the brickwork removed to show the various drums and connections. The construction of this boiler is very similar to that of the one just described. There is a nest of inclined tubes, the ends of which are expanded into cast-iron headers. The headers are placed in communication by the U shaped return bends a, a. A continuous channel is, therefore, provided for the circulation of the water through the headers. There is a horizontal overhead drum A for each vertical section of tubes. These drums A, A are placed in communication with the transverse drum B by the tubes C, C. The drum B is in turn connected with the lower drum H by the two large water legs D, D. Finally the drum H communicates with the rear headers through the tubes L, L. There is thus an open circuit through the tubes T, drum A, tubes C, drum B, water legs D, drum H, and tubes L. The water-line is at about the middle of the drums A, A, and the steam arising from the surface of the water first passes into the drum P, and then into the main steam drum S through the pipes Q, Q. The

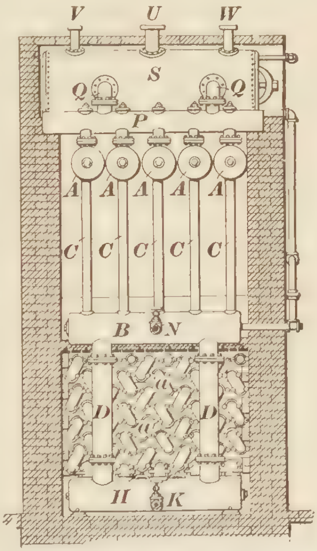


FIG. 503.

main steam pipe, the safety valve, and other fittings may be attached to the drum *S* at the nozzles *U*, *V*, and *W*.

The feed-water is introduced into the drum *B* through the feed-pipe *N*. The circulation takes place in the same manner as in the boiler just described. The drum *H* acts as the mud drum, being at the lowest point of the water circuit. The sediment may be blown out through the pipe *K*.

Access may be had to the interior of the setting through the doors *F*, *F*. The steam drum is provided with a manhole. The rear end of the boiler is supported by the brickwork foundation; the front end is supported by a beam *G* hung from the I beam *E*.

The arrangement of the bridge and baffle plates *J*, *J* and the course of the heated gases are precisely the same as in the Babcock and Wilcox boiler.

The points of superiority claimed by the makers of the Root boiler are: the flexible construction of the headers, allowing for expansion and contraction; the use of numerous small overhead drums instead of one or more large ones; the introduction of the feed-water into a separate drum; the admission of the circulating water directly into the lower tubes, thus protecting them from the intense heat of the furnace.

There are several other makes of water-tube boilers, varying only in detail from the two just described. They all consist essentially of a bank of 4-inch tubes, inclined at an angle of about 15 degrees, connected by headers and tubes with a horizontal steam drum, the tubes, headers, and drums forming a closed circuit through which the water circulates.

1712. The **Heine water-tube boiler,** shown in Fig. 504, differs in many respects from those already described. It consists of a large main drum *A* which is above and parallel with the nest of tubes *T*, *T*. Both drum and tubes are inclined at an angle with the horizontal that brings the water-level to about $\frac{1}{3}$ the height of the drum in front and about $\frac{2}{3}$ the height in the rear. The ends of the tubes are expanded into the large wrought-iron water legs *B*, *B*.

These legs are flanged and riveted to the shell, which is cut out for about $\frac{1}{4}$ of its circumference to receive them, the opening being from 60 to 90 per cent. of the cross-sectional area of the tubes. The drum heads are of a hemispherical form, and, therefore, do not need bracing.

The water legs form the natural support of the boiler, the front water leg being placed on a pair of cast-iron columns E which form part of the boiler front, while the rear water leg rests on rollers (shown at F) which may move freely on

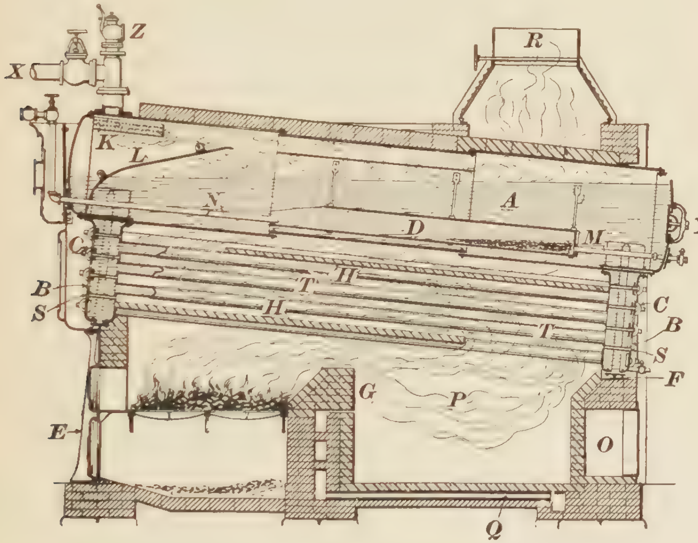


FIG. 504.

a cast-iron plate bedded in the rear wall. These rollers allow the boiler to expand freely when heated.

The boiler is enclosed by a brickwork setting in the usual manner. The bridge G, made largely of firebrick, is hollow, and has openings in the rear to allow air to pass into the chamber P and mix with the furnace gases. This air is drawn from the outside through the channel Q in the side wall, and it is, of course, heated in passing through the bridge. In the rear wall is the arched opening O which is closed by a door and further protected by a thin wall of firebrick,

which may be removed when it is necessary to enter the chamber P, and afterwards replaced.

The feed-water is brought in through the feed-pipe N, which passes through the front head. As the water enters, it flows into the mud drum D which is suspended in the main drum below the water line, and is thus completely submerged in the hottest water in the boiler. This high temperature is useful in precipitating the impurities contained in the feed-water. These impurities settle in the mud drum D and may then be blown out through the blow-out pipe M.

Layers of firebrick H, H act as baffle plates, and force the furnace gases to pass back and forth between the tubes. The gases finally escape through the chimney R placed above the rear end of the boiler. The drum in the vicinity of the chimney is protected by firebrick, as shown in the figure, to protect the steam space from the action of the hot gases.

The steam is collected and freed from water by the perforated dry pipe K. The main steam pipe with its stop valve is shown at X, the safety valve at Z. In order to prevent a combined spray of mixed water and steam from spurting up from the front header and entering the dry pipe, a deflecting plate L is placed in the front end of the drum.

A manhole Y is placed in the rear head of the drum. The flat sides of the water legs, which are made hollow to give access to the outside of the tubes, are stayed together by the staybolts S, S. A handhole C is placed in front of each tube to give access to its interior.

Where a battery of several of these boilers is used, an additional steam drum is placed above and at right angles to the drums A, A.

1713. The **Stirling boiler,** shown in Fig. 505, is a departure from the regular type of water-tube boilers. It consists of a lower drum A connected with three upper drums B, B, B by three sets of nearly vertical tubes. These upper drums are in communication through the curved tubes C, C, C. The curved forms of the different sets of

tubes allow the different parts of the boiler to expand and contract freely without strain.

The boiler is enclosed, as shown, in a brickwork setting, which is provided with various holes *H*, *H*, so that the interior may be inspected or repaired. The boiler is suspended from a framework of wrought-iron girders not shown in the figure.

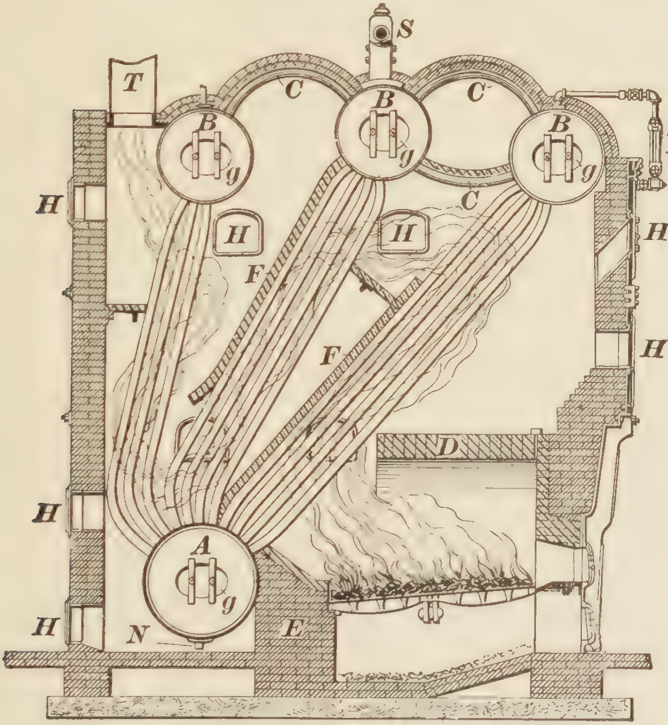


FIG. 505.

The bridge *E* is lined with firebrick, and is built in contact with the lower drum *A* and the front nest of vertical tubes. An arch *D* is built above the furnace, and this, in connection with the bafflers *F*, *F*, directs the course of the heated gases, causing them to pass up and down between the tubes. The arch and bafflers are made of firebrick.

The cold feed-water enters the rear upper drum and descends through the rear nest of tubes to the drum *A*, which acts as a mud drum, and collects the sediment brought in by the water. A blow-off pipe *N* permits the removal of the sediment. The steam collects in the upper drums *B*, *B*. The steam pipe and safety valve *S* are attached to the middle drum.

The chimney *T* is located behind the rear upper drum. Therefore, the cold feed-water enters the coolest part of the boiler, and the circulation of the water is directly opposite to that of the escaping hot gases.

The water column *L*, with its fittings, is placed in communication with the front upper drum. All the drums are provided with large manholes *g*.

The boiler is made with a cast-iron front.

The following advantages are claimed for the Stirling boiler :

(1) The vertical position of the tubes prevents the collection of sediment, and at the same time encourages the rapid rise and separation of the steam as soon as it is formed. (2) The boiler is very simple and easy to construct ; there are no flat surfaces to be stayed, and there is little or no machine work required in its manufacture. (3) It is very accessible for cleaning or repairs; any part of the boiler may be inspected by removing the four manhole plates *g*.

The various water-tube boilers just described are coming into extensive use. The most important points in their favor are their safety from disastrous explosion and their economy in the use of fuel. An objection sometimes urged against water-tube boilers is that they require more attention; since they usually have much less cubic capacity than cylindrical boilers of the same power, the water-level must be closely watched.

1714. The **Harrison safety boiler,** shown in Fig. 506, is a sectional safety boiler, but not of the water-tube type. It is composed of hollow cast-iron or steel sections *A*, *A*, called *units*, which are accurately faced and bolted

together. Each section is composed of two or more approximately spherical vessels, and the sections are bolted together in a zigzag manner, as shown in the figure, so as to form a solid slab. Each bolt runs from top to bottom through all the units as shown at *C*. There are several of these vertical slabs of sections suspended side by side from the girders *B*, *B*.

The boiler is enclosed by brickwork setting, which is lined with firebrick. The top of the boiler is likewise covered with firebrick to prevent radiation. The wall is pierced with the openings *D D* for the purpose of inspecting the

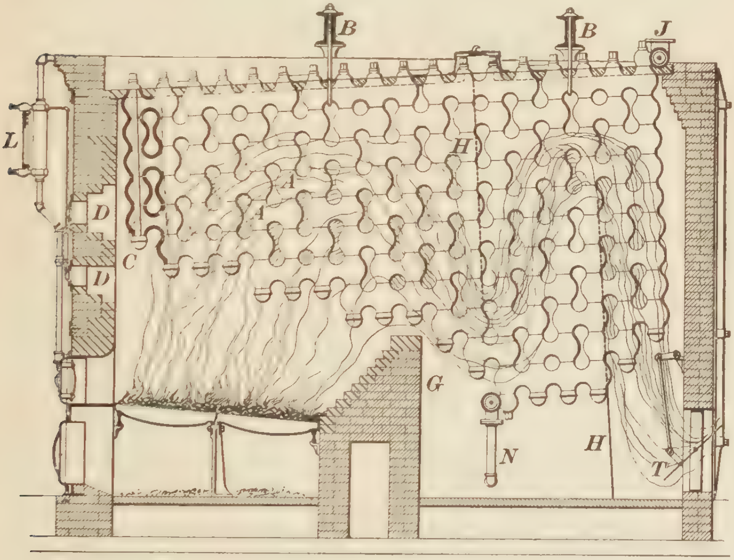


FIG. 506.

interior. The bridge *G* and bafflers *H H* direct the hot gases back and forth between the sections.

The feed-water enters the boiler at its lowest point through the feed-pipe *N*. The steam pipe is bolted on at the flange *J*. The water column *L*, placed in front at the height of the water-level, is connected by pipes with the steam and water spaces, respectively.

The chimney flue is placed at the rear near the floor; the draft is regulated by the damper *T*.

The boiler is essentially a safety boiler. If the steam pressure becomes excessive, the bolts C will elongate a little and allow steam to escape through the joints. Even if the pressure should be sufficient to burst a unit there would be no disastrous explosion, and it would be only necessary to replace the unit.

1715. The **Hazelton** or **porcupine boiler,** shown in Fig. 507, is a vertical water-tube boiler of peculiar form. As shown in the figure, it consists of an upright cylinder A, into which are expanded a large number of radial tubes B, B, whose outer ends are closed, while the inner ends open into the cylinder A. The boiler is enclosed in a circular brickwork wall, on top of which is placed the chimney H, provided with a damper for regulating the draft. Below the tubes the wall is lined with firebrick and projects inwards, forming the furnace E. In one side of the furnace is placed the fire door K. The grate G forms a ring between the central cylinder and the external brick wall. The space below the grate serves as the ash pit. The water is contained in the cylinder and tubes. The sediment naturally collects in the bottom of the cylinder and is blown out through the pipe D or removed through the man-hole C. The steam is collected in the perforated dry pipe P and

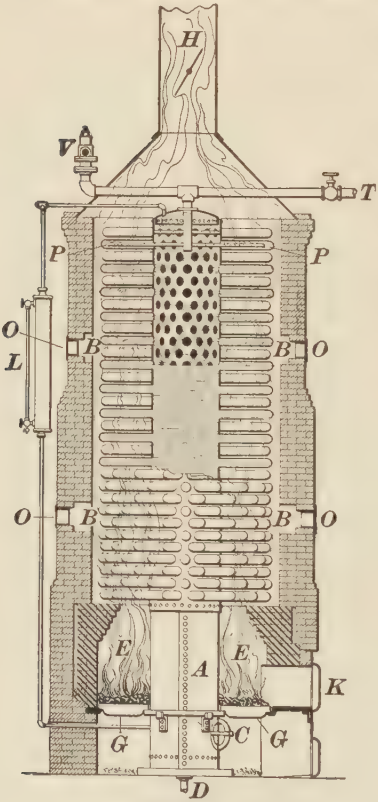


FIG. 507.

led to the main steam pipe *T*. The safety valve is shown at *V* and the water column at *L*. The openings *O*, *O* are left in the wall, so that the interior may be inspected.

The heated gases pass from the furnace *E* between the tubes *B*, *B*, and by the time they have reached the chimney, the heat has been mostly absorbed by the water in the tubes.

This boiler does not require to be suspended in any way; the whole weight rests upon the foundation, which may be built in the ground.

In addition to the types of boilers above described, there are many others embodying special features, or of peculiar construction. Some of these are modifications of the types already described; others have nothing in common with them.

1716. Of these miscellaneous types, the **Field boiler** may be mentioned. This boiler consists of a vertical cylindrical shell containing a plain flat-top fire-box. A single large flue passes from the crown sheet of the fire-box through the upper head of the boiler shell to the chimney. The remarkable feature of this boiler is the use of the so-called "Field tubes." The construction of one of these tubes is shown in Fig. 508. It consists of two concentric tubes, the outer one being closed at one end and open at the other, which is expanded into the crown sheet *C*; the closed end hangs down into the fire-box in contact with the hot furnace gases. The inner tube *B* is open at both ends and is suspended inside the outer tube by pins or feathers. The upper end of the inner tube is expanded so as to give a free entrance to the water.

FIG. 508.

The tubes being suspended in the fire-box, and exposed to intense heat, there is a rapid

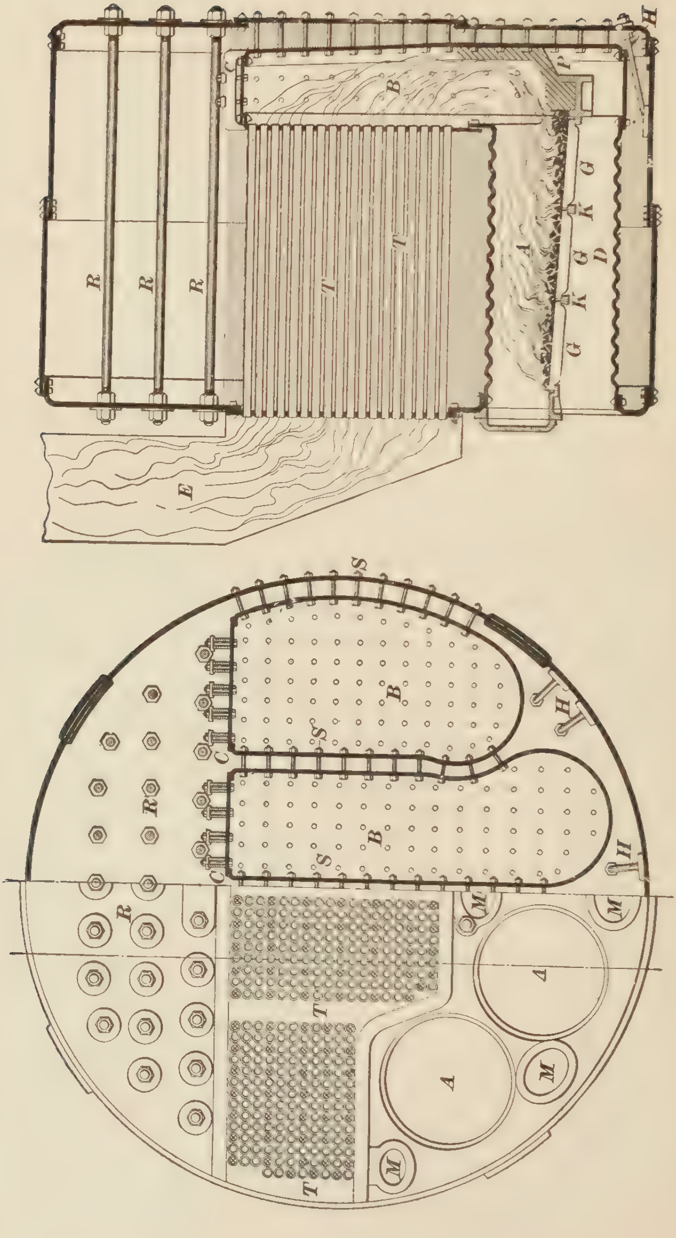


FIG. 509.

formation of steam which rises to the surface through the space between the outer and inner tubes. A stream of water is thus kept continually flowing downwards through the inner tube to supply the place of the rising steam, and the result is a rapid and continual circulation within the tubes.

Field tubes are also used in other boilers. Sometimes they are hung in the furnace flues of Cornish and Lancashire boilers. They are also used in some forms of fire-engine boilers.

MARINE BOILERS.

1717. The many types of marine boilers previously in use have been abandoned, and at the present time the majority of ocean steamers are fitted with the "Scotch" or "drum" boiler. Very recently, however, there has been a tendency to use different forms of water-tube boilers, especially for naval vessels and torpedo boats, and it is quite likely that in the future the Scotch boiler will be replaced to a greater or less extent by some form of water-tube boiler.

1718. The **Scotch** or **drum boiler,** shown in Fig. 509, has a cylindrical shell with flat heads. These boilers may vary from 10 to 15, or even 20, feet in diameter and from 7 to 11 feet in length. They are provided with two, three, or four large corrugated furnace flues *A*, *A* (the one shown in the figure has four), the rear end of which opens into a combustion chamber *B*. Usually each flue has its own combustion chamber, but in some cases two or more flues open into a common chamber. A nest of tubes *T*, *T* extends from the front plate of the combustion chamber to the front head of the shell. These flues place the combustion chambers *B*, *B* in communication with the large smoke chamber *E*, which in turn leads directly to the stack.

The flat heads of the shell are kept from bulging by the heavy stayrods *R*, *R*, and further by the diagonal braces *H*, *H*. About one-third of the tubes (those marked with a cross in the figure) are threaded and provided with nuts, and

thus act as stayrods for the flat surfaces occupied by the tubes. The flat sides of the combustion chambers are stayed to each other and to the rear head by the staybolts *S*, *S*. The flat tops of the combustion chambers, called the **crown sheets,** are strengthened by the girder stays, or **crown bars,** *C*, *C*. The manholes *M*, *M* give access to the various parts of the boiler. The various fittings are not shown in the figure, but are attached in convenient places.

The furnaces are placed within the corrugated flues *A*, *A*. As shown, the grate *G* is made in three sections, supported by the cross-bars *K*, *K*. Below the grate is the ash pit *D*. At the rear end of the grate is placed a firebrick plate *P* for the purpose of preventing cold air from sweeping through the ash pit into the combustion chamber without first passing through the grate.

The gases arising from the combustion of the coal pass into the combustion chamber *B*, where they undergo further combustion in contact with the air which passes through the grate. The hot products of combustion then pass through the tubes to the smokebox *E* and out through the stack.

It is seen from the figure that the flues and tubes are completely surrounded by water; likewise the combustion chambers. It is evident, therefore, that this type of boiler has a very large heating surface in proportion to its cubic contents.

1719. In Fig. 510 is shown a section of a **double-ended Scotch marine boiler.** At the left end of the figure the section is taken through one of the lower flues, and at the right end, through one of the upper flues. The furnace flues are placed in each end of the boiler, the combustion chamber, being near the center. Each chamber is connected with the nearest head by a nest of tubes leading to the smoke flues *H*, *H*.

The flat surfaces are stayed and braced as above described for the single-ended boiler.

The process of combustion and the path of the gases are

the same as for the boiler shown in Fig. 509. The plates *P*, *P* are to prevent air from passing from the ash pit to the

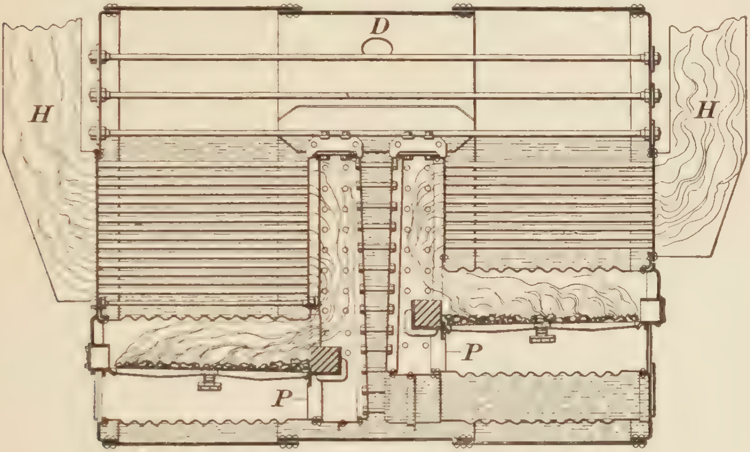


FIG. 510.

combustion chamber. The steam dome or drum is attached at *D*.

1720. Another form of double-ended boiler is shown in

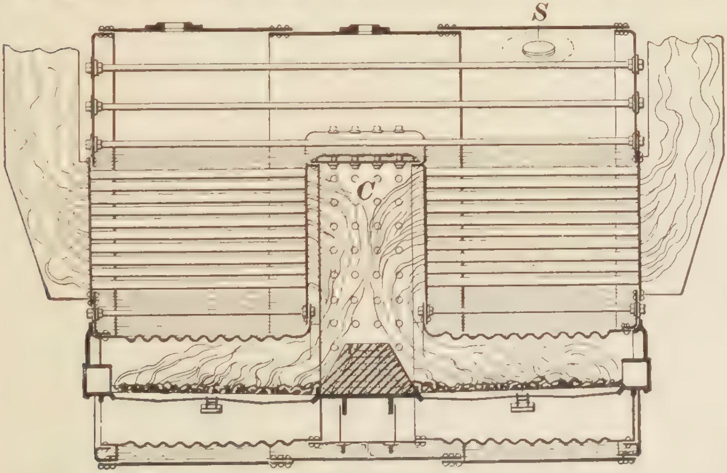


FIG. 511.

Fig. 511. The furnace flues are placed in each end of the shell, but each opposite pair opens into a common combus-

tion chamber *C*. Each of these combustion chambers has two nests of tubes, one nest connecting it with one head, the other nest with the other head. The gases from two opposite furnaces mix together in the common combustion chamber, and then pass through the two nests of tubes, one-half to one smoke flue, the other half to the other. In other respects the construction of the boiler is similar to that shown in Fig. 510. The steam dome is attached at *S*.

1721. On account of the high steam pressures used by modern marine engines, the marine boiler must be carefully designed for strength. It is likewise necessary to reduce its weight and size to the lowest possible limits. The following data concerning the boilers of a naval vessel will give an idea of the principal dimensions of a Scotch marine boiler:

Boilers of Siemens-Martin steel—

Diameter of shell	15 ft. 2″
Length of shell	9 ft. 6″
Working pressure	135 lb. per sq. in.
Thickness of shell plates	$1\frac{5}{32}$″
Thickness of heads	$\frac{7}{8}$″
Number of furnace flues	4
Diameter of furnace flues	3 ft. 1″
Thickness of furnace flues	$\frac{15}{32}$″
Diameter of stayrods	$1\frac{15}{16}$″
Diameter of staybolts	$1\frac{1}{4}$″
Number of tubes	490
Diameter of tubes	$2\frac{1}{2}$″
Length of tubes	6 ft. 8″
Heating surface	2,500 sq. ft.
Weight without water	about 40 tons.

It is clear from the above table that, for the sake of safety, the Scotch type of boiler must be made extremely heavy and bulky when high steam pressures are used, and much attention is now being paid to the devising of a new type of boiler which, while retaining the good features of the Scotch type, will be lighter, smaller, and cheaper for the same power.

BOILER DETAILS.

BOILER MATERIALS.

1722. The **materials** used in boiler construction are *wrought iron*, *steel*, *cast iron*, and, to some extent, *copper* and *brass*. The qualities required of boiler materials are: 1. Ductility, in order that the boiler may endure changes of form without rupture. 2. Tensile strength, to resist the rupturing stresses to which the boiler is subjected. 3. Toughness and elasticity.

1723. **Wrought iron** was until a few years ago the only material of which boiler shells, fire-boxes, and tubes were made. It is usually specified that iron for shell plates shall be of the best quality, fibrous, free from lamination, cracks, or blisters, and shall have a certain tensile strength, usually 50,000 to 55,000 pounds per square inch. As a measure of ductility, good boiler iron should show an elongation of at least 20 per cent. in a length of 8 inches, that is, when a bar 8 inches long is placed in a testing machine, it should stretch $8 \times \frac{20}{100} = 1\frac{6}{10}$ inches before breaking. Iron used for rivets should be of the very best quality; the tensile strength should be nearly or quite 60,000 lb. per sq. in., and a good rivet should bend double while cold without fracture.

1724. **Steel** is now rapidly supplanting iron as a boiler material. It exceeds iron in tensile strength, thus permitting the use of thinner sheets; it is more ductile and more homogeneous than iron, and by the modern processes it can be manufactured more cheaply than iron. The steel used in boiler construction belongs to the class known as soft or mild steel. It should contain less than $\frac{1}{4}$ of one per cent. carbon, and as little sulphur and phosphorus as possible. The tensile strength should not be much above 65,000 lb., for though higher strengths may be easily obtained, the plates will be harder and less ductile.

1725. The following are the specifications for the steel used in the boilers of the U. S. Navy: For shell plates the tensile strength shall be between 58,000 and 67,000 pounds per square inch; elongation, 22 per cent. in a length of 8 inches. For plates requiring to be flanged, the required tensile strength is 50,000 to 58,000 lb.; elongation 25% in 8 inches. Chemical requirements: Phosphorus, not over .035 of 1%; sulphur, not over .04 of 1%. Cold bending test: Specimen to stand being beat flat upon itself. Quenching test: Steel heated to cherry red, plunged in water at 82° F., and to be bent around curve $1\frac{1}{2}$ times the thickness of the plate.

Steel boiler plate is generally made by the Siemens-Martin process, though some is produced by the Bessemer process.

Greater care must be exercised in working steel than in working iron. Punching and shearing are apt to injure steel plates, and they must sometimes be annealed after these operations.

1726. Cast iron is used only in sectional boilers like the Harrison (shown in Fig. 506), and in these cases, cast steel is often substituted for it. Cast iron has some advantages as a boiler material. It is cheap, durable, and will withstand corrosion. Its brittle and treacherous nature, however, prevents its use in cylindrical boilers, except for mountings and settings. Sometimes, however, the ends or heads of plain cylindrical and flue boilers are made of heavy plates of cast iron.

1727. Copper is used in England for locomotive fireboxes and staybolts. Iron and steel are used for this purpose in the United States.

1728. Brass was formerly used in the construction of tubes. At present it is only used in the construction of some of the fittings.

STRENGTH OF BOILER SHELLS AND FLUES.

1729. Formula **111**, Art. **1365**, may be used for calculating the strength of boilers and flues subjected to internal fluid pressure. Thus, letting S_1 be the ultimate

tensile strength of the material, and substituting it for S, the above mentioned formula becomes $p\,d = 2\,t\,S_1$. Representing the safe working stress by $\frac{S_1}{f}$, in which $f =$ the factor of safety, we have, $p\,d = \frac{2\,t\,S_1}{f}$.

1730. When designing boilers, the tensile strength is specified, the factor of safety, diameter, and pressure per sq. in. are decided upon, and the only quantity remaining unknown is the thickness t. Solving the foregoing equation for t gives,

$$t = \frac{f\,p\,d}{2\,S_1}. \qquad \textbf{(164.)}$$

All plates used for making boiler shells should be tested and have their ultimate tensile strength stamped upon them. The lowest value stamped on the plate used should be substituted for S_1 in formula **164.** For single-riveted joints, the factor of safety f is taken as 6 by the United States Government, and for double-riveted joints, as 5.

1731. Again, the riveted joints of a shell have only from 50% to 80% of the strength of the solid plate. This percentage is called the efficiency of the joint; denote it by y. It is evident that to obtain the true strength of the shell the strength of the weakest part, that is, the riveted joint, should be calculated, or, what is the same thing, the quantity S_1 in formula **164** should be multiplied by the efficiency y. Formula **164** then becomes

$$t = \frac{f\,p\,d}{2\,S_1\,y}. \qquad \textbf{(165.)}$$

Should it be required to find the pressure per sq. in. which a given boiler will carry, solve formulas **164** and **165** for p. Hence,

$$p = \frac{2\,t\,S_1}{f\,d}; \qquad \textbf{(166.)}$$

$$p = \frac{2\,t\,S_1\,y}{f\,d}. \qquad \textbf{(167.)}$$

EXAMPLE.—Find the safe steam pressure, which may be used in a boiler 44″ in diameter and $\frac{5}{16}$″ in thickness, the efficiency of the joint being 64 per cent., the ultimate tensile strength 55,000 pounds, and the factor of safety 5.

SOLUTION.—By formula **167**,

$$p = \frac{2\,t\,S_1\,y}{f\,d} = \frac{2 \times \frac{5}{16} \times 55{,}000 \times .64}{5 \times 44} = 100 \text{ lb. per sq. in.} \quad \text{Ans.}$$

EXAMPLE.—Required the thickness of a boiler shell which is 5 ft. in diameter, under a pressure of 90 lb. per sq. in. The shell is of steel of a tensile strength of 60,000 lb., the factor of safety 4, and the efficiency of the joint is assumed to be 72 per cent.

SOLUTION.—By formula **165**,

$$t = \frac{p\,d\,f}{2\,S_1\,y} = \frac{90 \times 5 \times 12 \times 4}{2 \times 60{,}000 \times .72} = \frac{1''}{4}. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. Assuming a factor of safety of 6, and an efficiency of joint of 55%, find the pressure which may be used in an iron boiler, 3 ft. 6 in. in diameter and $\frac{1}{4}$ in. thick, the tensile strength of the plates being 50,000 lb. per sq. in. Ans. 54.6 lb. per sq. in., nearly.

2. Calculate the necessary thickness of a shell 32″ in diameter, the safe working stress being 10,000 lb. per sq. in., and the efficiency of the joint 60 per cent. The working pressure is 120 lb. per sq. in.

Ans. $\frac{5}{16}$ in.

NOTE.—When the calculated thickness comes out as a decimal, take the nearest $\frac{1}{16}$. For example, if the calculated thickness is .32 inch, take $\frac{5}{16}$″ as the proper dimension.

3. The efficiency of the joint is 66 per cent., the tensile strength of the plate 54,000 lb., the steam pressure 80 lbs. per sq. in., the diameter of the shell 50 inches, and the thickness of the shell $\frac{5}{16}$ inch. What is the factor of safety of the boiler shell? Ans. 5.57.

4. Find the thickness of a shell 54 inches in diameter, if the steel plates have a tensile strength of 63,000 lb. The efficiency of the joint is 70 per cent., and 6 is to be taken as the factor of safety. Steam pressure is 150 lb. Ans. $\frac{9}{16}$ in.

5. Assuming the seam to have $\frac{5}{8}$ the strength of the plate, and allowing a factor of safety of 4, what is the maximum safe diameter of a boiler shell $\frac{3}{8}$″ thick, and under a pressure of 125 lb.? The tensile strength of the plate is 50,000 lb. Ans. 50 in.

6. Of two boiler shells made of the same material, one is 30 inches in diameter and $\frac{1}{4}$″ thick; the other is 48 inches in diameter and $\frac{3}{8}$″ thick. Which is the stronger, and in what ratio?

Ans. The first is stronger in the ratio of 16 : 15.

7. Compute the thickness of the shell of a marine boiler $13\frac{1}{2}$ ft. in diameter, under a steam pressure of 160 lb. The safe working stress of the steel is 12,000 lb., and the efficiency of the seam is 80 per cent.
Ans. $1\frac{3}{8}$ in.

8. What pressure may be safely carried by a welded water tube, 4 inches in diameter and $\frac{5}{32}''$ thick? The safe working stress is 11,000 lb. per sq. in.
Ans. $859\frac{3}{8}$ lb.

1732. Flues and tubes subjected to an external pressure fail by collapsing rather than by bursting. When a cylinder is subjected to internal pressure, as a boiler shell, the pressure, being equal in all directions, tends to maintain the vessel in a truly cylindrical shape. When, however, the vessel is subjected to *external* pressure, as in the case of flues, the pressure tends to distort the vessel from its true shape. Thus, supposing a cylinder to be slightly elliptical

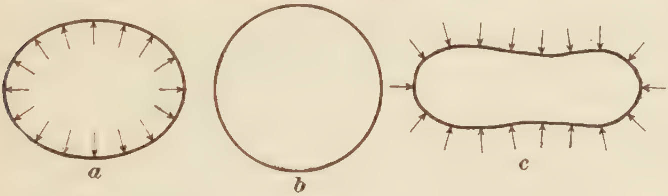


FIG. 512.

as shown at *a*, Fig. 512, an internal pressure would tend to return it to its original form, as shown at *b*, while an external pressure would tend to collapse it as shown at *c*.

The formulas for the strength and thickness of flues can not, therefore, be deduced by abstract reasoning like formulas **165** and **167**.

For this case use formula **114**, Art. **1368**, $p = \frac{9.600,000\,t^{2.18}}{l\,d}$. For all practical purposes, the square of the thickness may be used instead of the 2.18th power. Then, using a factor of safety of 6, the above formula becomes,

$$p = 1,600,000\,\frac{t^2}{l\,d}, \qquad \textbf{(168.)}$$

in which p = pressure in pounds per square inch, and all the other dimensions are in inches.

EXAMPLE.—What safe external pressure may be allowed on a pipe 14 ft. long, 3″ in diameter, and $\frac{5}{16}$″ thick ?

SOLUTION.—Applying formula **168,**

$$p = 1,600,000 \frac{(\frac{5}{16})^2}{(14 \times 12) \times 3} = 310 \text{ lb. pressure per sq. in.} \quad \text{Ans.}$$

RIVETING AND RIVETED JOINTS.

1733. Riveting is one of the most important operations in boiler making, for upon the character of the riveting depends in a large measure the strength and safety of the boiler. The flat plates as received from the rolling mill are first sheared or cut to the proper size, and the rough edges are planed off. The rivet holes are then punched or drilled in the edge of the plate; the plate is bent to the required shape and is then ready for riveting.

1734. The common forms and proportion of rivets are given in Figs. 513 to 515. In Fig. 513, the rivet before being upset is shown by the head and the dotted lines; it is upset or headed down to a cone, having a base double the diameter of the hole and a height $\frac{3}{4}$ the diameter of the hole. The rivet shown in Fig. 514 has a **snap** or **cup** head which is produced by hammering the rivet down roughly and finishing the head by a cup-shaped die or **set.** The heads in both these cases are produced by hand. The rivets shown in Figs. 515 and 516 are examples of machine riveting. The rivet is placed between two dies which are forced together by heavy steam or hydraulic pressure. When carefully done, machine riveting is much to be preferred to hand riveting.

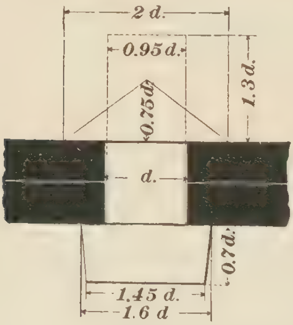


FIG. 513.

1735. A rivet with countersunk head is shown in Fig. 517. Such riveting is sometimes necessary where a smooth surface is needed to attach boiler mountings.

The proportional parts are given in terms of the diameter of the rivet hole. For example, the diameter of the countersunk head of Fig. 517 is 1.6 times the diameter of the rivet hole, and the diameter of the other head is 1.7 times the

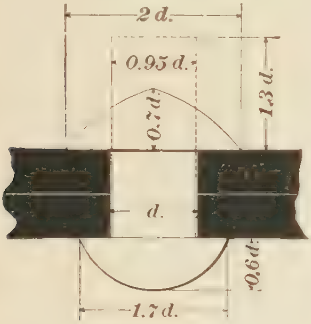


FIG. 514.

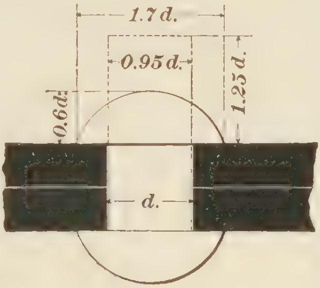


FIG. 515.

diameter of the hole. As shown in the figures, the rivets, before being headed, are slightly smaller than the hole, so

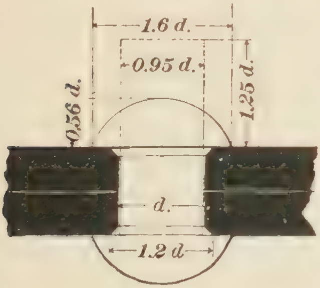


FIG. 516.

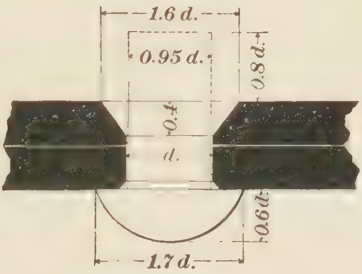


FIG. 517.

that they may be easily inserted. The upsetting action of heading the rivets causes them to fill the holes when headed down.

1736. **Riveted joints** of different forms are shown in Figs. 518 to 521. When one plate overlaps the other, and the two are joined with one or more lines of rivets, as shown in Figs. 518 and 519, the joint is called a **lap joint.** When, however, the plates are placed edge to edge, as in Figs. 520

and 521, and the joint is covered with one or two plates, the joint is called a **butt joint.**

1737. Fig. 518 represents a **single-riveted lap joint,** that is, the plates are overlapped and joined with one row of

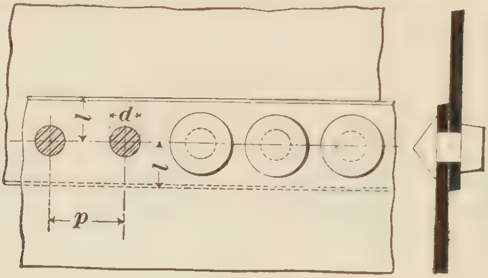


FIG. 518.

rivets. The distance *p* from center to center of the rivet holes is called the **pitch** of the rivets. The distance *l* from the center line of the rivet hole is usually made $1\frac{1}{2}$ times the diameter of the rivet hole *d*.

1738. Fig. 519 shows a **double-riveted lap joint.** The rivets may be **staggered,** as shown in the figure, or placed one behind the other, similar to those of Fig. 521. In the latter case the joint is **chain riveted.** It is quite customary in boiler construction to single rivet the girth seams and double rivet the longitudinal seams, since the latter are twice as liable to rupture as the former.

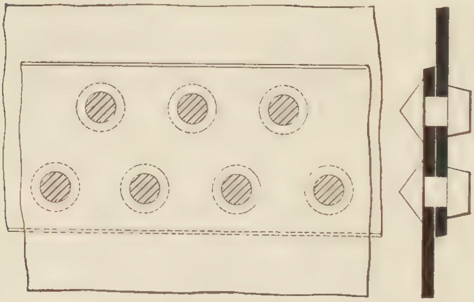


FIG. 519.

1739. A **butt joint** with a single cover plate is shown in Fig. 520, while a butt joint with two cover plates is shown

in Fig. 521. Either may be riveted with one, two, or more rows of rivets; Fig. 521 shows an example of chain riveting. The butt joint with two plates is stronger than one with a single plate, since the rivets tend to shear off in two places instead of one, and may, therefore, be of smaller diameter. In other words, the rivets in the former case have double the shearing surface

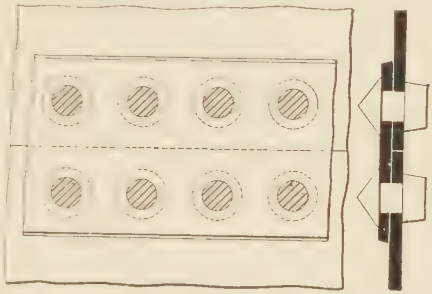


FIG. 520.

that they have in the latter. Butt joints are generally used for plates over $\frac{1}{2}$ inch thick, and are taking the place of lap joints in good designs of smaller work. When one cover plate is used on a butt joint, its thickness is $1\frac{1}{8}$ times

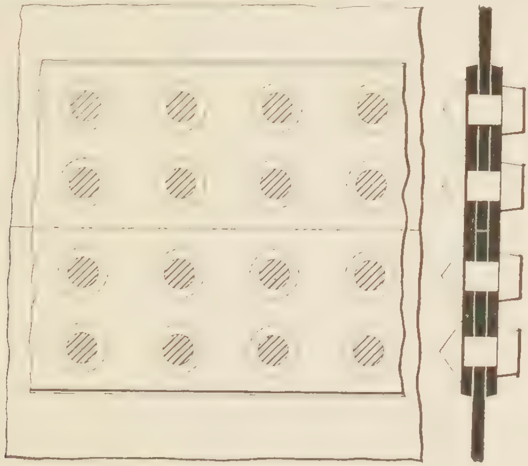


FIG. 521.

the thickness of the plate; when two cover plates are used, the thickness of each should be about $\frac{5}{8}$ of the plate thickness.

1740. Strength of Riveted Joints.—The first thing to consider is the manner in which a joint may fracture. (1) The plate may break along the line of the rivet

holes, as shown at *a*, Fig. 522. (2) The rivet may shear off, as shown at *b*, Fig. 522. (3) The plate in front of the rivet

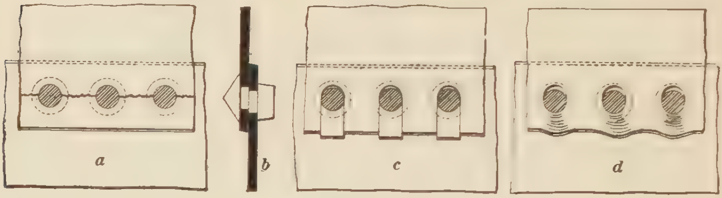


FIG. 522.

may shear out, as shown at *c*, Fig. 522. (4) The plate may crush in front of the rivet, as shown at *d*, Fig. 522.

Let $d =$ diameter of rivet hole in inches ;
$h =$ pitch of rivets in inches;
$t =$ thickness of plate in inches;
$f =$ factor of safety;
$n =$ number of rivets per foot of length of joint;
$l =$ distance from center of rivet to edge of plate in inches;
$S_1 =$ ultimate tensile strength per square inch of plate;
$S_2 =$ ultimate crushing strength per square inch of the plates.
$S_3 =$ ultimate shearing strength per square inch of rivets.

Consider a portion of a single-riveted joint 1 foot ($= 12$ inches) in length, then $n = \frac{12}{h}$, and let P be the total pressure tending to rupture that portion of the joint. The length of plate cut out by rivets is nd. Hence, the sectional area of the plate left (*a*, Fig. 522) is $(12 - nd)\,t$. Its resistance to rupture is $(12 - nd)\,t\,S_1$.

Therefore, $P = \frac{(12 - nd)\,t\,S_1}{f}$, or substituting for n its value $\frac{12}{h}$, we have

$$\frac{12\,t\,S_1}{h}(h - d) = Pf. \quad (a)$$

The shearing area of each rivet being $\frac{1}{4}\pi d^2$, the total shearing resistance of the rivets in a foot of length is $\frac{\pi}{4} d^2 n S_3$; therefore, $\frac{\pi}{4} d^2 n S_3 = \frac{\pi}{4} d^2 S_3 \times \frac{12}{h} =$

$$\frac{3 \pi d^2 S_3}{h} = Pf. \quad (b)$$

For the third case, the shearing area in front of the rivets is $2\,l\,t\,n$. Hence, the resistance to shearing is $2\,l\,t\,n\;S_3$, and, therefore, $2\,l\,t\,n\,S_3 = Pf$, or

$$\frac{24\,l\,t\,S_3}{h} = Pf. \quad (c)$$

For the case of crushing in front of the rivet hole (d, Fig. 522), the projected crushing area in front of each rivet is $d\,t$. Hence, the resistance to crushing is $d\,t\,n\,S_2$, consequently, $d\,t\,n\,S_2 = Pf$, or

$$\frac{12}{h} d\,t\,S_2 = Pf. \quad (d)$$

If the joint is double riveted, (b) and (d) become, respectively,

$$\frac{6 \pi d^2 S_3}{h} = Pf, \quad (b')$$

$$\frac{24\,d\,t\,S_2}{h} = Pf, \quad (d')$$

while (a) will remain the same. If a butt joint with double cover plates is used, the rivets are in double shear, and, hence, for single riveting (b) becomes

$$\frac{6 \pi d^2 S_3}{h} = Pf,$$

and for double riveting it becomes

$$\frac{12 \pi d^2 S_3}{h} = Pf.$$

In a properly designed joint the thickness t and distance l should be equal to or greater than that required by the above formula, and the pitch h should be equal to or less than the pitch required by (b).

1741. As an example of the use of the equation, let it be required to design the riveting and find the thickness of the

plate of a steel boiler, 3 ft. 9 in. in diameter, working under a steam pressure of 80 lb. Let $f = 6$; $S_1 = 54{,}000$; $S_2 = 90{,}000$, and $S_3 = 50{,}000$.

The reason for using such a high value for S_2 in this case is that experiment has shown that the crushing strength in front of the rivet is considerably higher than in the case of a single isolated piece of wrought iron, and that the average crushing strength per sq. in. of the area of the plate projected on the diameter is about 90,000 lb.

The total tension on a portion of a joint 12 in. long is,

$$P = \frac{45 \times 80 \times 12}{2} = 21{,}600 \text{ lb.}$$

Substituting in (b), $\frac{3\pi d^2 S_3}{h} = Pf$, we have

$$\frac{3\pi d^2 \times 50{,}000}{h} = 6 \times 21{,}600.$$

$$\frac{\pi d^2}{h} = \frac{6 \times 21{,}600}{3 \times 50{,}000} = .864.$$

Assume $d = \frac{3}{4}$, then $\frac{\pi(\frac{3}{4})^2}{h} = .864$,

or $h = 2$ in., nearly, which gives 6 rivets to the foot. Substituting these values of h and d in (a),

$$\frac{12\, t S_1}{2}(2 - \tfrac{3}{4}) = 6 \times 21{,}600,$$

$$\text{or } t = \frac{6 \times 21{,}600 \times 2}{12 \times 54{,}000 \times 1\frac{1}{4}} = .32 \text{ in.}$$

It will be safe to take $t = \frac{11}{32}$ in.

From equation (c),

$$l = \frac{fPh}{24\, t S_3} = \frac{6 \times 21{,}600 \times 2}{24 \times \frac{11}{32} \times 50{,}000} = .63 \text{ in.}$$

Hence, the usual custom of making $l = 1\frac{1}{2}d$ is perfectly safe in this case.

Finally, from equation (d),

$$\frac{12}{h} d t S_2 = \frac{12}{2} \times \frac{3}{4} \times \frac{11}{32} \times 90{,}000 = 139{,}218.75 \text{ lb.}$$

Therefore, the plate is safe against crushing in front of the rivets, since Pf only equals $21,600 \times 6 = 129,600$ lb.

The thickness of plate is, of course, found by equation (a), equation (d) being simply used as a check, to prevent the plate from being made too thin to resist crushing.

Combining (a) and (b) and assuming S_1 and S_3 to be equal, which is substantially true for wrought iron, we obtain

$$\frac{12\,t\,S_1}{h}(h-d) = \frac{3\,\pi\,d^2\,S_3}{h}.$$

Hence, $h - d = \frac{\pi d^2}{4t}$, and $h = d + \frac{\pi d^2}{4t} = \left(1 + .7854\frac{d}{t}\right)d.$

1742. For greater safety the decimal .7854 is increased in practice to $.875 = \frac{7}{8}$, and the pitch is found by the following formula:

$$h = \left(1 + \frac{7\,d}{8\,t}\right)d. \qquad \textbf{(169.)}$$

For double-riveted lap joints or single-butt joints, formula **169** becomes

$$h = \left(1 + \frac{7\,d}{4\,t}\right)d. \qquad \textbf{(170.)}$$

For double-riveted double-butt joints, formula **170** becomes

$$h = \left(1 + \frac{7\,d}{2\,t}\right)d. \qquad \textbf{(171.)}$$

The "d" in the above formulas refers to the diameter of the rivet hole, or of the rivet after being headed.

The diameter of the cold rivet is $\frac{1}{16}''$ less.

For single and double-rivet lap joints and butt joints, with one cover plate, the diameter of the rivet is

$$d = t + \tfrac{3}{8}'' \qquad \textbf{(172.)}$$

for values of t between $\frac{1}{4}''$ and $\frac{3}{4}''$. For butt joints with double cover plates, the diameter of rivet may be taken about $1\frac{1}{4}$ times the thickness of the plate.

1743. The following table gives the proportions of riveted joints used by some of the best boiler-making concerns in the United States:

TABLE 37.

Thickness of Plate. t.	Diameter of Rivet.	Diameter of Hole. d.	Pitch.		Efficiency of Joint.	
			Single.	Double.	Single.	Double.
$\frac{1}{4}''$	$\frac{5}{8}''$	$\frac{11}{16}''$	$2''$	$3''$	.66	.77
$\frac{5}{16}''$	$\frac{11}{16}''$	$\frac{3}{4}''$	$2\frac{1}{16}''$	$3\frac{1}{8}''$	.64	.76
$\frac{3}{8}''$	$\frac{3}{4}''$	$\frac{13}{16}''$	$2\frac{1}{8}''$	$3\frac{1}{4}''$	.62	.75
$\frac{7}{16}''$	$\frac{13}{16}''$	$\frac{7}{8}''$	$2\frac{3}{16}''$	$3\frac{3}{8}''$	.60	.74
$\frac{1}{2}''$	$\frac{7}{8}''$	$\frac{15}{16}''$	$2\frac{1}{4}''$	$3\frac{1}{2}''$	.58	.73

The preceding table is applicable to lap joints and butt joints with single-cover plate. For plates more than $\frac{1}{2}$ inch thick, double-covered butt joints are recommended.

The efficiencies in the above table are obtained from the formula

$$y = \frac{h - d}{h}. \qquad \textbf{(173.)}$$

For example, the efficiency of a double-riveted plate $\frac{7}{16}''$ thick is $y = \frac{h-d}{h} = \frac{3\frac{3}{8} - \frac{7}{8}}{3\frac{3}{8}} = .74$, as given above.

EXAMPLES FOR PRACTICE.

1. Find the proper pitch and diameter of rivet holes for a single-riveted lap joint, the plate thickness being $\frac{13}{32}''$.

Ans. $\begin{cases} h = 2''. \\ d = \frac{25}{32}''. \end{cases}$

2. Find the pitch and diameter of rivet holes for a double-riveted butt joint with two covers, the plate being $\frac{9}{16}''$ thick.

Ans. $\begin{cases} h = 3\frac{10}{16}'', \text{ nearly.} \\ d = \frac{11}{16}''. \end{cases}$

3. Let the student design the riveting for a shell 50 inches in diameter, 100 lb. steam pressure, by using equations (a), (b), (c), and (d). Assume the shell to be of steel, and let $f = 5$, $S = 50{,}000$, $S_1 = 60{,}000$, and $S_2 = 90{,}000$.

1744. Arrangement of Joints and Plates.—The plates of externally fired boilers should be so arranged that

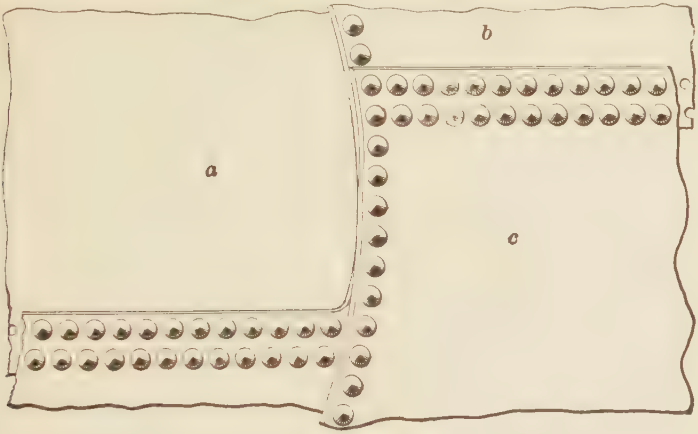


FIG. 523.

the riveted joints are as far as possible from the fire. This may be accomplished by using extra large plates for the furnace end of the shell.

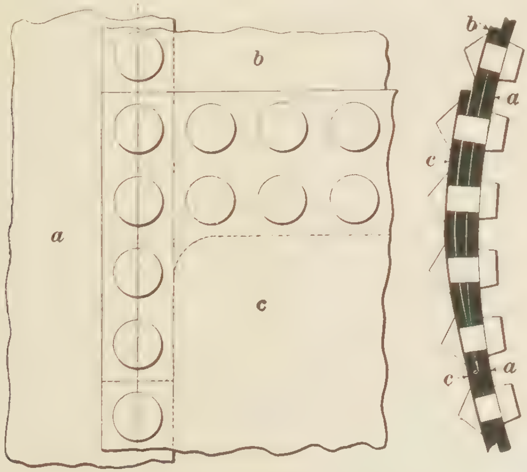


FIG. 524.

1745. Whenever a girth seam occurs, the longitudinal seams should break joint as shown in Fig. 523. In order to

make a tight joint where the three plates come together, the inner plate of the longitudinal joint must be hammered thin at the edge, as shown in Fig. 524.

In the construction of both vertical and horizontal shells, it is customary to have the inside lap facing *downwards*, since, if it faces upwards, a ledge is formed on which sediment may be deposited.

Since wrought-iron plates are stronger in the direction of the fiber, they should be arranged so that the fiber runs circumferentially around the shell; that is, in the direction of the girth seams.

1746. Connecting Plates.—Different methods of connecting plates at right angles are shown in Figs. 525 to

FIG. 525.

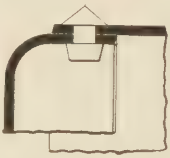


FIG. 526.

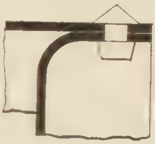


FIG. 527.

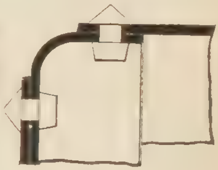


FIG. 528.

528. In Fig. 525 the two plates are riveted to an angle iron. This construction is used sometimes for connecting the heads of a boiler to the shell. Figs. 526 and 527 show the head flanged and riveted to the shell, while in Fig. 528 the head and shell are connected by a flanged ring.

Iron for flanging should be of the best quality. The radius of the curve to which the head is flanged should be at least 4 times the thickness of the plate.

Some makers of large marine boilers prefer to flange the end plates of the shell to receive the head, which is, consequently, a flat disk.

1747. In Figs. 529 to 535 is shown the usual construction of the water legs and furnace doors of vertical and fire-box boilers. Fig. 529 shows the door constructed by flanging the furnace sheet *A* and the front sheet of the boiler *B*. Single riveting is shown here, although it is frequently double riveted. An enlarged view of this construction is shown in Fig. 530. The door *C* is generally made of

cast iron, and is hinged to a cast-iron frame which is usually held in position by four studs $\frac{7}{8}''$ in diameter. Sometimes the frame is omitted and the door is made of wrought iron; it is then held in position by riveting the hinges to the boiler.

Around the lower ends of the water legs, or around the bottom of the furnace, and between the inside and outside

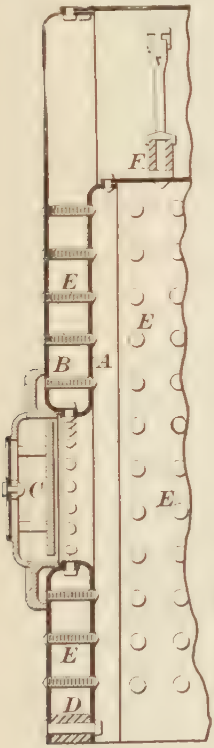


FIG. 529.

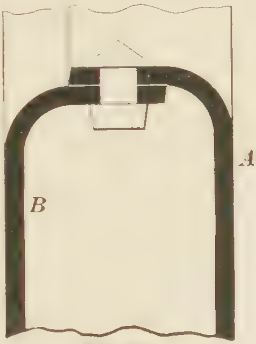


FIG. 530.

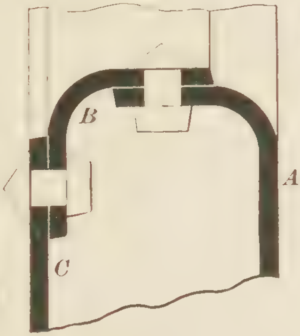


FIG. 532.

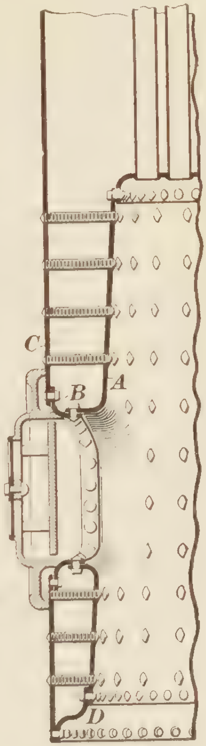


FIG. 531.

furnace plates, is riveted a wrought-iron ring D. In cheap boilers this ring is frequently made of cast iron. The stay-bolts E, E and crown bar F will be considered later. Instead of flanging both sheets in constructing the furnace door, as in Figs. 529 and 530, it is sometimes constructed as shown in Figs. 531 and 532. Here the furnace sheet A is

flanged, and to it is riveted the flanged ring *B* which has been riveted to the shell of the boiler *C*. An enlarged view of this construction is shown in Fig. 532. A flanged ring *D*, Fig. 531, is sometimes used at the bottom of the water leg in place of a wrought-iron ring *D*, Fig. 529, one of the

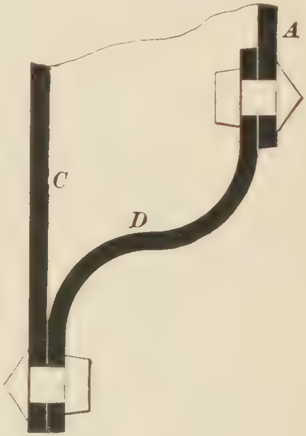


FIG. 533.

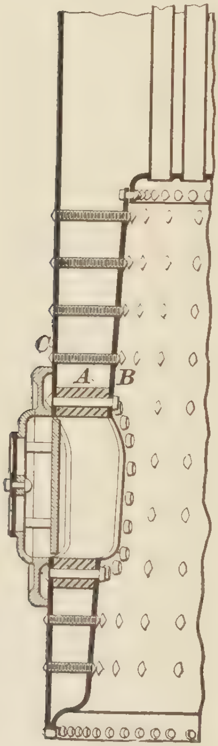


FIG. 534.

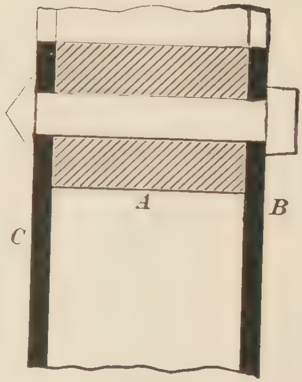


FIG. 535.

flanges being riveted to the furnace plate and the other to the shell, as shown. An enlarged view of this construction is shown in Fig. 533. In Fig. 534 is shown another method of constructing the furnace door and bottom of the water

leg. In this construction the wrought-iron ring A is placed between the furnace plate B and the shell of the boiler C, and riveted to them. An enlarged view of this construction is shown in Fig. 535. At the bottom of the water leg the furnace plate is flanged and riveted to the shell as shown.

1748. Rivet holes are either punched or drilled. Punching, while cheaper than drilling, is generally believed to injure the plates, particularly if they are at all hard or brittle. Many makers follow the practice of punching the hole smaller than its intended diameter and then reaming it out, which cuts away the injured metal around the holes. Annealing the plates after punching will partly remove the injury. It is the present practice of good boiler-makers to drill all steel plates and punch iron plates.

1749. Calking is an upsetting process applied to a riveted joint in order to make it steam tight. The operation is shown in Fig. 536. A round-nose calking tool is driven against the beveled edge of the upper plate, forcing the metal in close contact with the lower plate, and effectually closing the seam. A tool with a sharp edge should never be used, as it is liable to score the under plate and lead to grooving.

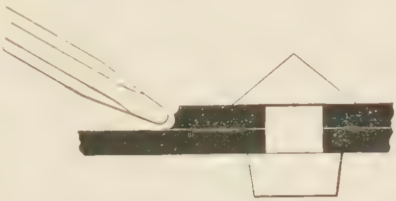


FIG. 536.

FLAT SURFACES—STAYING.

1750. The surfaces of boiler shells are in general either cylindrical, hemispherical, or flat. A cylinder or sphere subjected to an internal steam pressure is *self-supporting*; that is, the steam pressure tends to maintain the cylindrical or spherical form of the vessel, and hinders distortion instead of producing it. If, on the contrary, the vessel is composed of flat surfaces, the steam pressure tends to distort it and

give it an approximately spherical form. Hence, flat surfaces are not self-supporting, and must be braced or stayed to prevent distortion.

The flat surfaces commonly found in boiler construction are the flat heads of cylindrical shells and the flat sides and crown of the fire-box in the locomotive type.

1751. Strength of Flat Surfaces.—The strength of a circular plate supported at the edges, as, for example, the head of a boiler shell, is given by the following formula:

$$S = \frac{2 r^2 p}{3 t^2}, \qquad \textbf{(174.)}$$

where S represents the safe tensile strength of the material; r, the radius of the plate in inches; t, the thickness of the plate in inches, and p, the pressure in pounds per square inch.

Solving equation **174** for t, we obtain

$$t = r\sqrt{\frac{2p}{3S}}. \qquad \textbf{(175.)}$$

Suppose, for example, it is required to find the necessary thickness of an *unstayed* head of a boiler shell 3 ft. in diameter, under a steam pressure of 80 lb. Assuming a safe tensile strength of say 8,000 lb. per sq. in. and substituting in formula **175**, we obtain,

$$t = r\sqrt{\frac{2p}{3S}} = \frac{36}{2}\sqrt{\frac{2}{3} \times \frac{80}{8,000}} = 1.47 \text{ in., or } 1\tfrac{1}{2} \text{ in., nearly.}$$

The shell plates in the above case would need to be about $\frac{5}{16}''$ thick. The enormous thickness required by unstayed flat plates is very apparent, and it is plain that flat ends of moderate thickness should be well strengthened by stays.

1752. The following formula gives the strength of a square unstayed flat plate fixed along the edge (the length of side is a):

$$\left.\begin{aligned} S &= \frac{a^2 p}{4 t^2}, \\ \text{or } t &= \frac{a}{2}\sqrt{\frac{p}{S}}. \end{aligned}\right\} \qquad \textbf{(176.)}$$

As an example of the use of the formula, let it be required to find the thickness of a fire-box plate, unstayed, 40 in. square and exposed to a steam pressure of 100 lb. per sq. in. Take $S = 8{,}000$ lb. per sq. in.

SOLUTION.—

$$t = \frac{a}{2}\sqrt{\frac{p}{S}} = \frac{40}{2}\sqrt{\frac{100}{8{,}000}} = 2\tfrac{1}{4} \text{ inches, nearly. Ans.}$$

Such a thickness is, of course, out of the question. Hence, when plates of ordinary thickness are used, it is plain that they must be strongly stayed.

BOILER STAYS.

1753. The principal kinds of stays in common use are: (1) Direct stays which are placed at right angles to the flat surfaces which they support; (2) diagonal stays and gusset stays which support the flat surface by tying it to another surface inclined to the first; (3) girder stays which are placed above the flat surface and bolted to it at intervals.

When the length of the shell is short in comparison with its diameter, the flat ends or heads are connected by stay-rods, as shown in Fig. 496. Stayrods are made of wrought iron or steel, and each end is generally threaded to receive two nuts, as shown in Fig. 537. The washers *a* and *b* are used to strengthen the plate. Frequently only one washer is used, as in Fig. 509; it is then placed on the outside of the sheet. Sometimes these washers are increased in diameter and riveted to the plate.

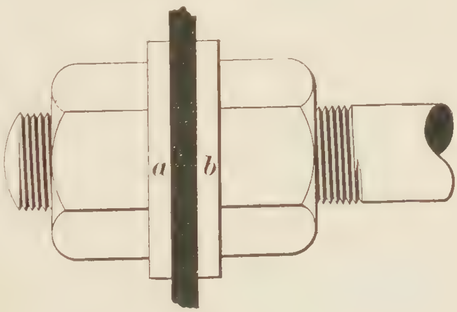


FIG. 537.

1754. Another method of connecting the ends of stayrods to the plate is shown in Fig. 538. Here the ends of the stayrod *A*, instead of being threaded, are forked to receive the connection *B*, which is riveted to the head of the boiler and joined to the stayrod by a bolt *C*. The combined effective area of the two legs of connection *A* should exceed the area of the stayrod.

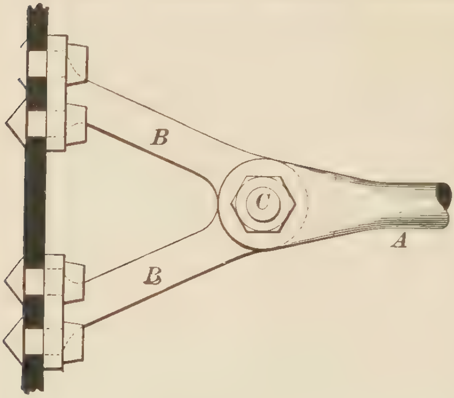


FIG. 538.

1755. Figs. 539, 540, and 541 represent different forms of staybolts used for strengthening the flat-side surfaces, and, in some instances, the circular surfaces of boilers. The one

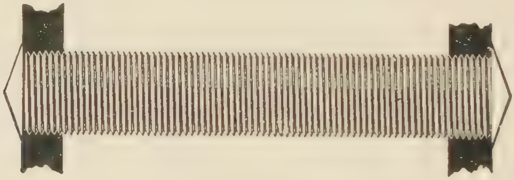


FIG. 539.

shown in Fig. 539 is generally used for staying the flat-side surfaces of fire-box boilers and the circular furnace sheets of vertical boilers, as will be seen by referring to Figs. 531, 533, and 536. The diameter of these bolts is either $\frac{3}{4}$ or $\frac{7}{8}$ of an inch. The one in Fig. 540 is used for the same purpose. Here the bolt is threaded only at the ends, the center being of the same diameter as the diameter at the bottom of the threads. In each end is drilled a small hole *a* for the purpose of detecting broken bolts, in which event steam will be seen issuing from the hole. The staybolt represented in Fig. 541 is generally used for staying the flat-side surface of marine boilers, as will be seen by referring to Figs. 509, 510,

and 511. Instead of riveting the ends, washers and nuts are used, the washers serving to strengthen the flat plates.

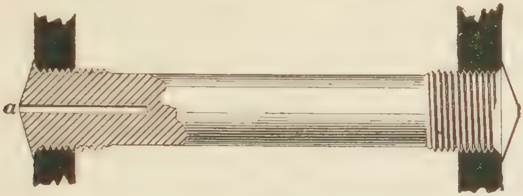


FIG. 540.

When these plates are of sufficient strength, the washers are dispensed with, and the nuts screwed close to the plates.

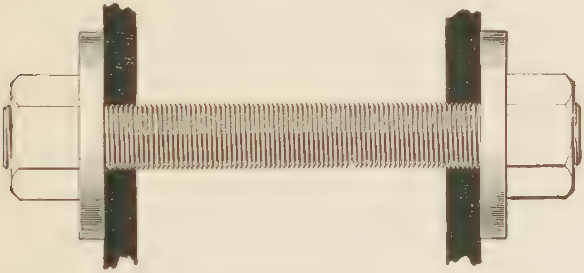


FIG. 541.

The staybolts used in marine boilers are of steel, and vary in diameter from $1\frac{1}{4}$ to $1\frac{5}{8}$ inches.

1756. The diameter of a direct stay may be found as follows:

Let A = area in square inches of plate supported by one stay;
d = smallest diameter of stay in inches;
p = pressure of steam in pounds per square inch;
T = safe stress in pounds per square inch allowed on stay.

Then, the total pressure on the stay is $A\,p$, and the resistance of the stay is $\frac{1}{4}\pi\,d^2\,T$. Hence,

$$\left.\begin{array}{l}\frac{1}{4}\pi\,d^2\,T = A\,p,\\ \text{or } d = 1.13\sqrt{\dfrac{A\,p}{T}}.\end{array}\right\} \qquad \textbf{(177.)}$$

Fire-box staybolts are generally arranged with the same pitch, both vertically and horizontally, as shown in Fig. 542. Denoting this pitch by a, it is evident that each

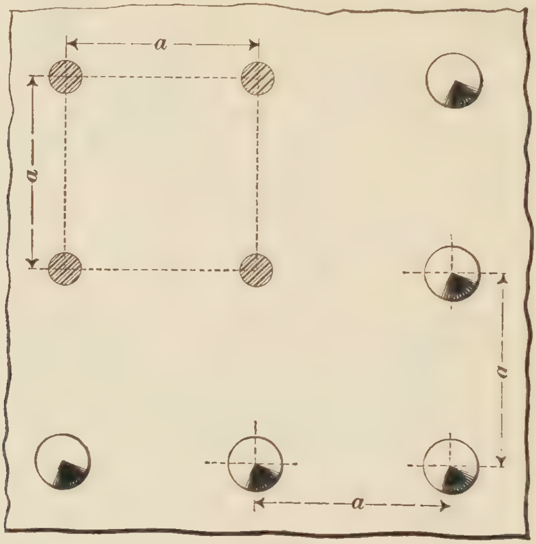


FIG. 542.

staybolt supports an area a^2 represented by the dotted lines. Hence, the above formula becomes $d = 1.13\sqrt{\frac{a^2 p}{T}}$, or

$$d = 1.13a\sqrt{\frac{p}{T}}, \qquad \textbf{(178.)}$$

$$\text{and } a = .885\,d\sqrt{\frac{T}{p}}. \qquad \textbf{(179.)}$$

1757. The values of T used in formulas **177, 178,** and **179** may be taken as follows:

For copper screw staybolts.......................... $T = 4{,}000$

Iron Stays.—For screw stays, and for other stays not exceeding $1\frac{1}{4}$ inches in diameter, and for all stays which are welded............... $T = 6{,}000$

For unwelded stays above $1\frac{1}{2}$ inches effective diameter.................................... $T = 7{,}500$

Steel Stays.—For screw stays, and for other stays not exceeding $1\frac{1}{2}$ inches effective diameter . . . . $T = 8{,}000$

For stays above $1\frac{1}{2}$ inches effective diameter . . . . . $T = 9{,}000$

No steel stays are to be welded.

EXAMPLE.—Find the effective diameter of the steel stayrods of a marine boiler, the rods being spaced 14 inches each way. Steam pressure 135 lb. per sq. in.

SOLUTION.—Using formula **178**, we obtain

$$d = 1.13\,a\sqrt{\frac{p}{T}} = 1.13 \times 14 \times \sqrt{\frac{135}{9{,}000}} = 1\tfrac{15}{16} \text{ in. Ans.}$$

EXAMPLE.—The copper staybolts of a locomotive fire-box have a diameter of $\frac{7}{8}$ in. The steam pressure is 150 lb. per sq. in. Determine the proper pitch of the stays.

SOLUTION.—By formula **179**,

$$a = .885\,d\sqrt{\frac{T}{p}} = .885 \times \frac{7}{8} \times \sqrt{\frac{4{,}000}{150}} = 4 \text{ inches. Ans.}$$

1758. Diagonal Stays.—When the boiler shell is long in proportion to its diameter, through stayrods are seldom used owing to the great lengths required, and, therefore, diagonal and gusset stays have been adopted. The diagonal or crowfoot brace is shown in Fig. 543. Here the end A is either bolted or riveted to the head, and the end B riveted to the shell.

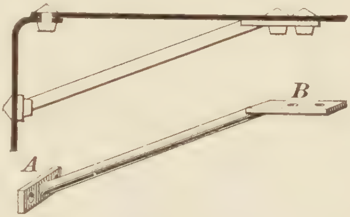


FIG. 543.

In marine boilers, where this style of stayrod is used, instead of the end A being riveted to the head, it is threaded and supplied with nuts and taper washers on each side of the head, as shown in Fig. 544, the washers having such a taper that when one of the faces is against the head, the other is parallel to the faces of the nuts. The hole through which the stay

FIG. 544.

passes is not threaded, but is made sufficiently large to allow the stay to pass through.

The combined area of the rivets attaching the stay to the shell should equal the area of the rod. The angle which a diagonal stay makes with the shell should not exceed 30°, and should be as much smaller as possible.

The size of a diagonal stay may be calculated from formula **178,** by substituting $\frac{p}{\cos B}$ for p, B being the angle between the stay and the shell. Formula **178** thus becomes

$$d = 1.13\sqrt{\frac{Ap}{T\cos B}}. \qquad \textbf{(180.)}$$

EXAMPLE.—(*a*) Find the diameter of a crowfoot brace which supports an area $6\frac{1}{2}''$ by $8''$ against a steam pressure of 90 lb. per sq. in. The angle B is 25° and the stress T is assumed to be 7,000 lb. (*b*) If the rod is fastened at each end by two rivets, find the diameter of the rivets.

SOLUTION.—(*a*) By formula **180,**

$$d = 1.13\sqrt{\frac{Ap}{T\cos B}} = 1.13\sqrt{\frac{6\frac{1}{2} \times 8 \times 90}{7{,}000 \times \cos 25^\circ}} = .97 \text{ in., say 1 in.} \quad \text{Ans.}$$

(*b*) Since there are two rivets, the area of each must be $\frac{1}{2}$ the area of the rod, or $\frac{1^2 \times .7854}{2} = .3927$ sq. in. The diameter is, therefore, .7 inch or $\frac{11}{16}$ in., nearly. Ans.

In the above formula, cos B may be easily obtained by dividing the horizontal distance from the inside of the head to the point where the stay is riveted to the shell, by the length of the stay.

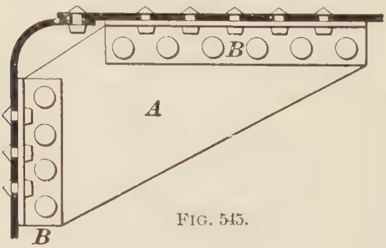


FIG. 545.

1759. In Fig. 545 is shown a gusset stay. These stays consist of a wrought iron or steel plate A, secured to the head and shell by either angle or T irons B, B. They are much used for staying the heads of internally fired boilers of the Lancashire and Galloway type. (See H, Fig. 496.)

In Figs. 546, 547, and 548 are shown sections of gusset

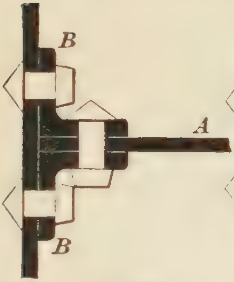


FIG. 546.

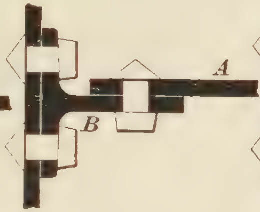


FIG. 547.

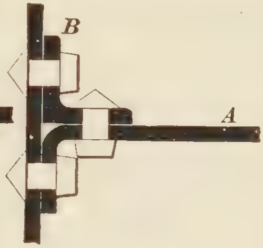


FIG. 548.

stays. Fig. 546 shows a method of using two angle irons with a plate between them; Fig. 547, a method of using a T iron with a plate riveted to one side, and Fig. 548, a method in which the gusset plate is flanged to one side and the angle iron riveted to the other. Gusset stays are placed radially in a boiler, the largest one in the center, and smaller ones to the right and left of it, as shown in Fig. 549.

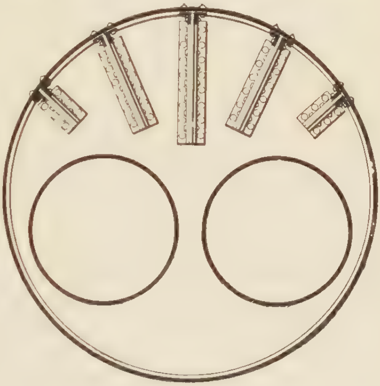


FIG. 549.

1760. Girder Stays, or Crown Bars.—It was previously stated that the side furnace plates of fire-box boilers, the side plates of combustion chambers in marine boilers, and the circular furnace plates of vertical boilers were strengthened with staybolts; but as yet nothing has been said regarding the staying of the upper plates or crown sheets. For strengthening these sheets in fire-box and marine boilers, girder stays, or crown bars, are used; in vertical boilers no special means are applied, since the tubes themselves furnish all the staying that is necessary. See Figs. 498 and 499. In Fig. 550 is shown the general stay-

ing of the crown sheet of a fire-box boiler in which A is a girder stay, or crown bar, resting partly on the crown sheet C and partly on the side furnace plates B, B to which the

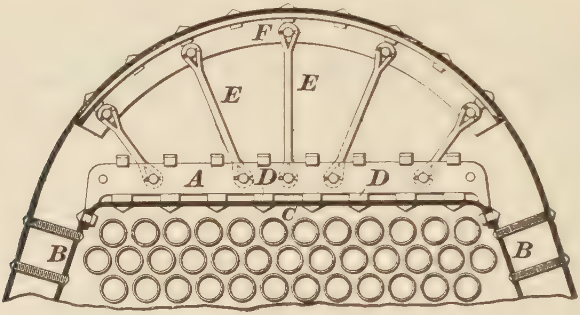


FIG. 550.

crown sheet is riveted. D, D are distance pieces used to prevent the crown sheet from bulging upwards when upsetting the rivets. Fig. 551 shows the general construction of the crown bars. Here two wrought-iron bars A and B are held apart by distance pieces C, C at both ends and riveted. The rivets used for staying the crown

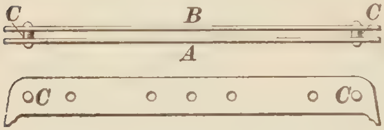


FIG. 551.

sheets of fire-box boilers to the crown bars are of wrought iron, and generally made like the one shown at D, Fig. 552. The projections D, D on the head are to prevent the bars from spreading. In staying the crown sheets of the combustion chambers of marine boilers, bolts are used instead of rivets. The bolts are made of steel, and provided with nuts on each end, as shown at

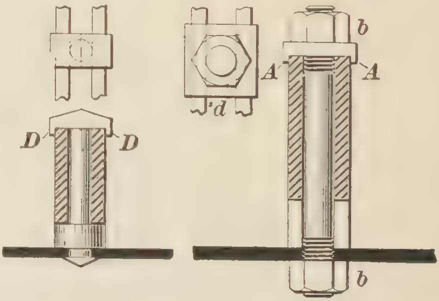


FIG. 552.

b, b, Fig. 552, the plate or washer d with lugs A, A and a hole in the center through which the bolt passes, being used

to prevent the bars from spreading. In fire-box boilers the crown sheet is additionally strengthened by the stayrods E, E, as shown in Fig. 550, one of the ends of the rods being secured to the crown bar by pins, and the other to a T iron F, which in turn is riveted to the shell of the boiler. These rods not only give additional strength to the crown sheet, but also brace the cylindrical part of the shell. Rivets used for staying crown sheets are usually made from $\frac{3}{4}$ to $\frac{7}{8}$ of an inch in diameter, and the bolts $1\frac{1}{4}$ inches.

1761. The dimensions of the girder may be found by considering it to be a simple beam uniformly loaded and supported at the ends.

Let l = length of girder in inches;
h = distance between adjacent girders in inches;
p = steam pressure in pounds per square inch;
b = breadth of girder in inches;
d = depth of girder in inches;
S_4 = ultimate transverse strength of girder in pounds per square inch;
f = factor of safety.

From the Table of Bending Moments, the bending moment on the above beam is $\frac{W l}{8}$. But $W = p\,h\,l$; hence, bending moment $= \frac{p\,h\,l^2}{8}$, and the moment of resistance is $\frac{S_4 I}{f c} = \frac{S_4 b d^2}{6 f}$; consequently, $\frac{p\,h\,l^2}{8} = \frac{S_4 b d^2}{6 f}$. Since the crown sheet itself forms part of the beam, though not taken into consideration in the above discussion, the high value of 18,000 lb. may be used for $\frac{S_4}{f}$; it is also customary to make $b = \frac{1}{4} d$. Substituting these values for b and $\frac{S}{f}$ in the above equation, and solving for d, we obtain

$$d = \sqrt[3]{\frac{p\,h\,l^2}{6{,}000}}, \qquad \textbf{(181.)}$$

$$\text{and } p = \frac{6{,}000\, d^3}{h\,l^2}. \qquad \textbf{(182.)}$$

EXAMPLE.—The girder stays on the combustion chamber of a marine boiler are 30 inches long and spaced 9 inches apart; the steam pressure being 150 lb., find the depth of the girder.

SOLUTION.—Using formula **181**,

$$d = \sqrt[3]{\frac{p\,h\,l^2}{6{,}000}} = \sqrt[3]{\frac{150 \times 9 \times 30^2}{6{,}000}} = \sqrt[3]{202.5} = 5\tfrac{7}{8} \text{ in. Ans.}$$

The width of each of the bars forming the girder is $\frac{1}{2} \times \frac{d}{4}$ $= \frac{5\frac{7}{8}}{8} = \frac{47}{64}$, or $\frac{3}{4}$ in., nearly.

1762. Flat surfaces which are not exposed to a very high pressure are sometimes stayed by simply riveting T irons or channel irons to the surface. This method is inefficient, and should be avoided.

1763. Strength of Stayed Surfaces.—Stays are designed to withstand the whole of the pressure coming on the flat surface, no allowance being made for the strength of the plate. Still, the heavy pressure tends to bulge and rupture the segments of the plate included between the stays, and the plate thickness must be sufficient to withstand it. The following formula, which is derived from experiments made under the direction of the U. S. Navy, may be used to compute the safe pressure allowable on a flat surface, strengthened by screw staybolts:

Let p = pressure in pounds per square inch;
h = pitch of staybolts;
t = thickness of plate;
k = a constant.

$$\text{Then, } p = k\frac{t^2}{h^2}, \qquad \textbf{(183.)}$$

$$\text{and } t = h\sqrt{\frac{p}{k}}. \qquad \textbf{(184.)}$$

The constant k has the following values:

For iron plates and iron bolts.................... $k = 24{,}000$
For low steel plates and iron bolts.............. $k = 25{,}000$
For low steel plates and low steel bolts.......... $k = 28{,}000$
For iron plates and iron bolts with nuts.......... $k = 40{,}000$
For copper plates and iron bolts.................. $k = 14{,}500$

EXAMPLE.—What safe pressure may be allowed on the fire-box plates of a locomotive boiler, the thickness of the plates being $\frac{1}{2}$ inch, and the pitch of stays being 4 inches? The plates and bolts are of iron.

SOLUTION.—Using formula **183**,

$$p = k\frac{t^2}{p^2} = 24{,}000 \times \frac{\left(\frac{1}{2}\right)^2}{4^2} = 375 \text{ lb.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. What should be the diameter of fire-box steel staybolts spaced $4\frac{1}{2}$ inches apart, the steam pressure being 160 lb.? Ans. $\frac{23}{32}$ in.

2. Find the proper diameter of wrought-iron welded stayrods, pitched 12 inches apart each way and sustaining a steam pressure of 120 lb. per sq. in. Ans. $1\frac{15}{16}$ in.

3. Determine the pitch of the screw staybolts of a locomotive fire-box, the bolts being of wrought iron, $\frac{7}{8}''$ in diameter, and the steam pressure 150 lb. per sq. in. Ans. $4\frac{7}{8}$ in.

4. Find the diameter of an unwelded wrought-iron diagonal brace which supports an area of 45 square inches against a pressure of 80 lb. per sq. in. The angle which the brace makes with the shell is 20°. Ans. .9 in.

5. The steel screw staybolts of combustion chambers of a marine boiler are pitched 7 inches apart. The pressure is 135 lb. per sq. in. What is the diameter of the bolts? Ans. 1 in., nearly.

6. What should be the depth of the girders of a locomotive fire-box, the girders being 30 inches long, spaced 5 inches apart? The steam pressure is 150 lb. Ans. $4\frac{7}{8}$ in., nearly.

7. What pressure may be allowed on a flat surface, the staybolts of which are pitched 8 inches apart? The plate is of iron $\frac{1}{2}$ inch thick, and the bolts are of iron with nuts. Ans. $156\frac{1}{4}$ lb. per sq. in.

8. Calculate the thickness of the wall of a steel fire-box subjected to a steam pressure of 145 lb. per sq. in., stayed with steel bolts, pitched $4\frac{1}{2}$ inches apart. Ans. $\frac{11}{32}$ in., nearly.

9. What would be the necessary thickness of an unstayed iron plate 42 inches square, riveted at the edges, assuming $S = 7{,}000$ lb. and the pressure 110 lb. per sq. in.? Ans. $2\frac{5}{8}$ in.

10. Required the thickness of the plate in example 9, on the supposition that it is stayed with iron bolts pitched 4 inches apart. Ans. $\frac{9}{32}$ in., nearly.

BOILER FITTINGS.

1764. The Feed Apparatus.—Water is supplied to a boiler either by a steam pump, by an injector, or by both. For the sake of safety every boiler should have two independent feeds, in order to prevent accident, should one get out of order.

The position of the feed-water pipe, and likewise the point where the feed-water discharges into the boiler, have been shown in the cuts of some of the boilers described. The feed-water pipe may enter the boiler either through one of the heads or through the shell. By some engineers it is placed in the front head directly over the furnace sheet of cylindrical, flue, or return tubular boilers; also, directly over the crown sheet of the furnace in vertical boilers, and in the head (through the smokebox) as low as possible in fire-box boilers; while others place it in the shell on top of the cylindrical, flue, return tubular, and fire-box boilers, and in the upper head in case of vertical boilers. Still others place it

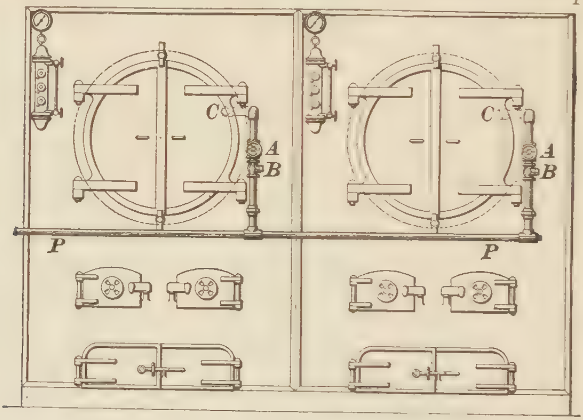


FIG. 553.

as low as possible in the back head of the cylindrical, flue, or return tubular boilers, or through the front or back head, just below the water line, near the shell. It is not good practice, however, to let the cold feed-water be delivered near the hot furnace plates, as the strains set up by the sudden cooling of the plates may seriously injure them. Feed-pipes should not terminate immediately at the plate into which they are screwed, but between the center of the boiler and the rear head. Sometimes two or three feet of the end of the feed-pipe are perforated for the purpose of diffusing the feed-water.

1765. In Fig. 553 is shown an ordinary method of arranging the feed-water pipes, where several boilers are

supplied by the same pump. The main pipe *P P*, running along the fronts of the boilers, receives the feed-water discharged from the pump. Each boiler is supplied by a pipe branching from the pipe *P*, and entering the front head *C*. Each of these branches is provided with a globe valve *A* and a check-valve *B*. The globe valve shuts off the water from the boiler, while the check-valve allows the water to enter the boiler when the globe valve is open, but prevents its return.

1766. The ordinary **globe valve** is shown in Fig. 554. The water enters at *A*, flows through the opening beneath the valve and out through *B*. It will be noticed that the valve seat is flat instead of being beveled. Flat-seated valves are commonly used, although perhaps not quite so good as those with beveled seats.

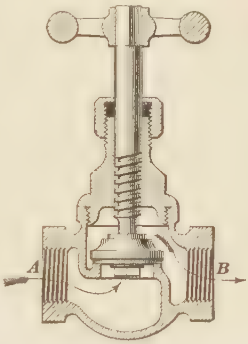


FIG. 554.

1767. The construction of a **swing check-valve** is shown in Fig. 555. It is evident that the water entering below the valve at *A* will raise it from its seat and flow through the outlet *B*, into the globe valve, and from thence into the boiler. The projection *C* on the valve strikes the bottom of the screw and is thus prevented from going too far. As soon as the pump ceases working, the pressure on the back of the valve forces it to its seat, and thus prevents the water from returning. The check-valve shown in the illustration is better suited for a horizontal than a vertical position.

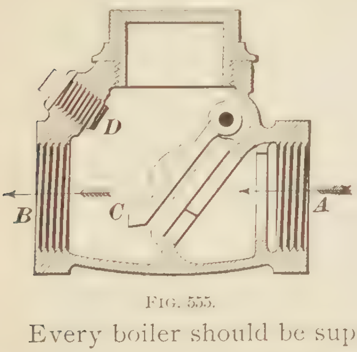


FIG. 555.

Every boiler should be supplied with its own independent check-valve, as otherwise there is danger that the steam pressure may force the water back through the feed-pipe.

The capacity of the pump or injector, whichever is used, should be considerably greater than is demanded in regular work. In this way the pump may be run very slowly and continuously for ordinary work, which is better both for pump and boiler, and there will be a reserve capacity for emergencies.

1768. Feed-Water Heaters.—It is important that the feed-water should be introduced into the boiler at as high a temperature as possible. The advantages of hot feed-water are: (1) The avoidance of the strains produced in the plates by the introduction of cold feed-water; (2) the saving in fuel effected by the higher temperature of the feed. The economy of using hot feed-water may be shown by a simple calculation.

Suppose a boiler to furnish steam at 75 lb. pressure, and, in the first case, let the temperature of the feed-water be 60° F. The number of heat units required to change a pound of water at 60° F. into steam at 75 lb. pressure is, from the steam tables, about 1,151 B. T. U. Suppose, in the second case, the feed-water enters at a temperature of 210° F. Then, the number of heat units gained by heating the feed is $210 - 60 = 150$ B. T. U., and the gain per cent. is $\frac{150}{1,151} = 13$ per cent.

1769. Feed-water heaters are of two classes: (1) Those which make use of exhaust steam from the engine; (2) those which make use of the waste furnace gases.

Heaters of the first class somewhat resemble the surface condenser in construction; they usually consist of a vessel, generally of cylindrical form, filled with rows or coils of tubes.

In some heaters the steam passes through the tubes which are surrounded by the feed-water. In other heaters the water is pumped through the tubes, which are, in this case, surrounded by the exhaust steam.

1770. A common form of feed-water heater is shown in Fig. 556, which shows two views—a longitudinal section through the shell and a vertical section through the inlet

feed-pipe *F*. It will be seen that the heater consists of an outer cylindrical shell and of an inner one fitted with numerous tubes. The feed-water enters through *F* and fills the space in the inner shell not occupied by the tubes. The exhaust

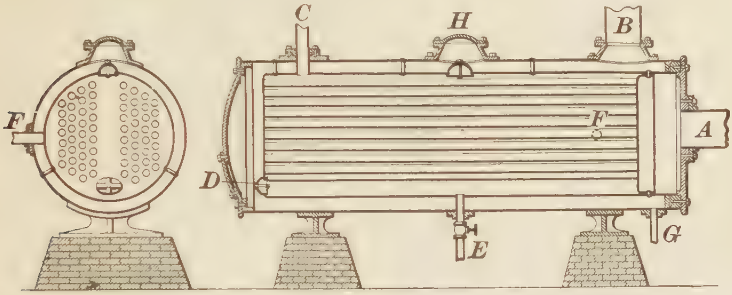


FIG. 556.

steam enters at *A*, flows through the tubes, then back through the space between the inner and outer shells and out through *B*. The feed-water flows out through *C* into the boiler. *D* is a handhole; *E*, the blow-off pipe, and *H*, a manhole.

When it is desired to economize space, vertical feed-water heaters are used instead of the horizontal pattern shown in the cut.

1771. Economizers make use of the heat in the waste furnace gases to raise the temperature of the feed-water. The temperature of the gases on entering the chamber is usually from 450° to 650° F., and by lowering this temperature to 250° or 300° a marked saving of fuel must result. The draft of the chimney, however, depends upon the temperature of the gases. The loss in draft consequent on the reduction of temperature may be made up by increasing the height of the chimney.

1772. Fig. 557 shows the location of an economizer with respect to the boilers and chimney. As will be seen, it is placed directly in the flue. The water enters at *F* where the economizer is coolest, and flows along the pipe *G*. From *G*, the water flows out at right angles to its former

direction through a series of horizontal radiating headers *K*, *K*, etc., and up the rows of vertical tubes *H*, *H*, etc.,

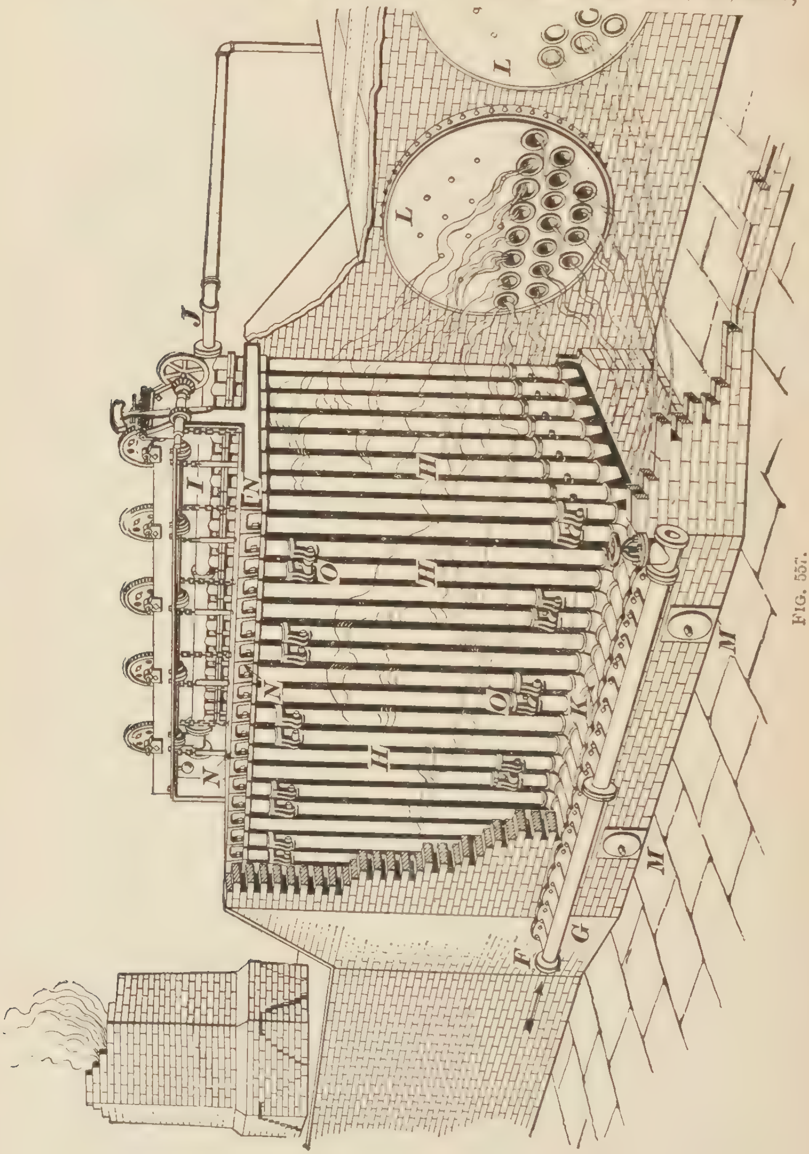


FIG. 557.

which connect with them. Each of these vertical rows has an upper header *N*, *N*, etc., which has one outlet into the delivery pipe *I*, to which is connected the pipe *J*, leading to the boilers *L*, *L*. The hot gases from the boilers pass through the rows of tubes on their way to the chimney, coming in contact with the rows containing the hottest water first. The feed-water may be heated by this means to as high as 300° F., and the temperature of the gases reduced from the neighborhood of 600° to 250° or 300°.

The hot gases deposit soot and other unconsumed particles upon the tubes. Since soot is a very bad conductor of heat, the efficiency of the economizer would soon be greatly impaired unless means were provided for removing the soot. This is accomplished by scrapers *O*, *O*, etc., which are moved up and down by means of a suitable mechanism on top of the economizer. *M*, *M* are openings for the removal of the soot scraped from the tubes.

SAFETY VALVES.

1773. A **safety valve** is attached to a boiler to prevent the steam from rising above its safe working pressure. When steam is made more rapidly than it is used, its pressure must of necessity rise; and if no means of escape is provided for it the result must be an explosion.

1774. The **dead-weight safety valve**, shown in

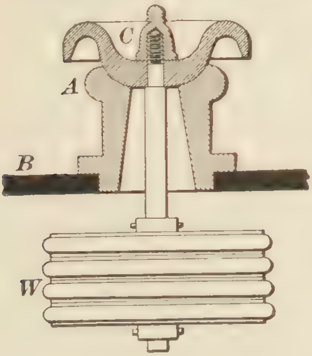


FIG. 558.

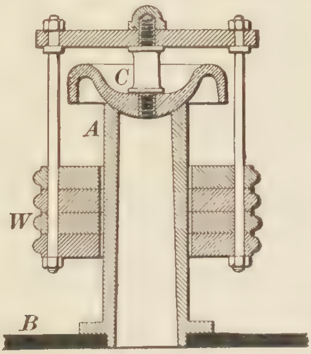


FIG. 559.

Fig. 558, consists of a hollow seat *A* attached to the boiler

shell B, over which is fitted the valve disk C. The disk is loaded with a heavy weight W which hangs down into the steam space of the boiler. Fig. 559 shows another form of dead-weight valve, in which the load is carried outside the boiler shell.

Let A = area of opening in valve seat in square inches;
p = pressure at which the valve is to blow off (in pounds per square inch);
W = dead weight in pounds (including weight of valve).

As soon as the total pressure on the valve is slightly in excess of the dead weight, the valve is lifted and the steam escapes. Therefore, the dead weight must equal the pressure per sq. in. multiplied by the area of the opening; that is, $W = A\,p$.

1775. Two forms of lever safety valves are shown in Figs. 560 and 561. The valve V is held to its seat

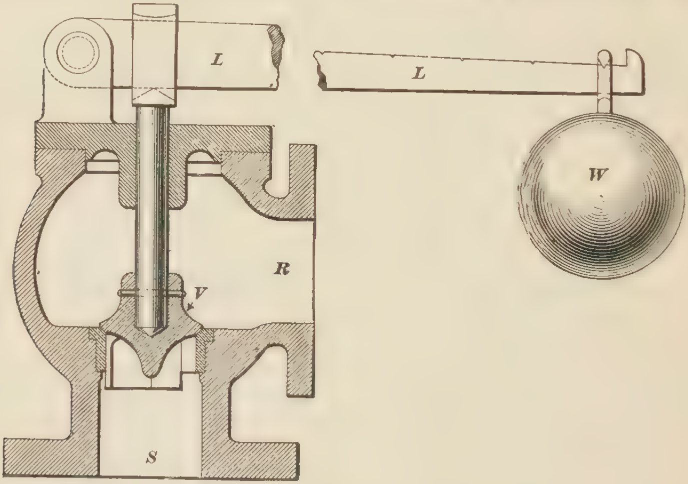


FIG. 560.

by the weighted lever L. The position of the weight W is adjustable, so that the valve may be set to blow off at different steam pressures. The valve shown in Fig. 560 is attached directly to the boiler shell; the steam enters from the boiler

at S and is discharged through the orifice R. The valve shown in Fig. 561 differs from the other in being attached to the supply pipe. The steam passes on its way from the boiler through the passage $S\ O$. When the pressure rises

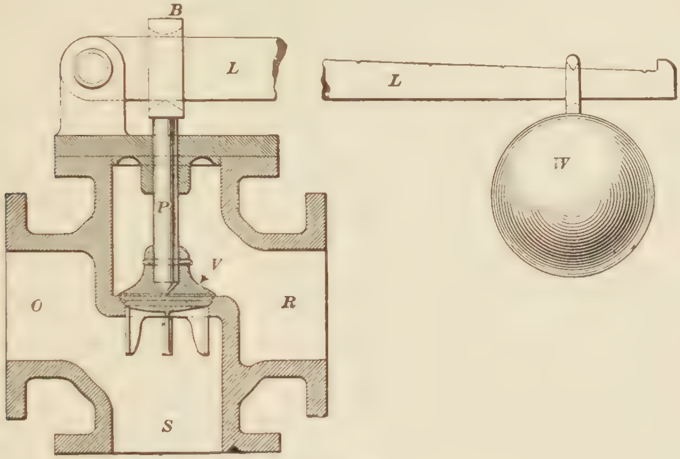


FIG. 561.

above the normal pressure, the valve V opens and the steam escapes into the air through the opening R.

1776. In the skeleton diagram, Fig. 562, the valve

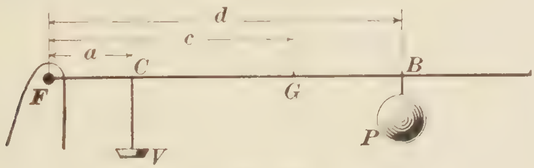


FIG. 562.

stem and weight are attached to the lever at C and B, respectively; the fulcrum is at F.

Let $d = F\,B =$ distance from fulcrum to weight in inches;
$c = F\,G =$ distance from fulcrum to center of gravity of lever in inches;
$a = F\,V =$ distance from fulcrum to center of valve in inches;
$A =$ area of orifice beneath bottom of valve in inches;

W = weight of weight P in pounds;
W_1 = weight of valve and stem in pounds;
W_2 = weight of lever in pounds;
P = blow-off pressure in pounds per square inch.

According to the rule given in Art. **1380,** it is a necessary condition of equilibrium that the algebraic sum of the moments of all the forces about a given point shall equal zero. Hence, taking F as the center of moments and treating all the forces acting downwards as positive and the upward pressure of the steam as negative, we have, since the upward steam pressure is $p\,A$,

$$W d + W_1 a + W_2 c - p\,A\,a = 0, \text{ or}$$

$$W = \frac{a(p\,A - W_1) - W_2 c}{d}, \qquad \textbf{(185.)}$$

$$\text{and } d = \frac{a(p\,A - W_1) - W_2 c}{W}. \qquad \textbf{(186.)}$$

1777. From these formulas may be obtained the required weight W, when its distance from the fulcrum is known, or *vice versa;* the point of attachment may be calculated when the weight is known.

EXAMPLE.—The area of the orifice is 10 sq. in., the distance from the valve to the fulcrum is 3 in., and the length of the lever is 32 in. The valve and stem weigh 5 lb., the lever weighs 12 lb., and the steam pressure is 90 lbs. What should be the weight W, if placed 2 in. from the end of the lever, assuming the lever to be straight?

SOLUTION.—Substituting in formula **185,**

$W = \dfrac{3(90 \times 10 - 5) - 12 \times 16}{30} = 83.1$ lb. Ans.; since $c = \dfrac{32}{2} = 16$ in. for this case.

Having decided upon the weight to be used, formula **186** will give the distances from the fulcrum at which the weight may be placed in order to allow the boiler to blow off at different steam pressures.

EXAMPLE.—Suppose all of the quantities to remain the same as in the solution of the last example, except that it is desired to have the boiler blow off at 75 lb. instead of 90 lb., what will be the distance of the weight from the fulcrum?

SOLUTION.—Applying formula **186,**

$$d = \frac{3(75 \times 10 - 5) - 12 \times 16}{83.1} = 24.58 \text{ in. Ans.}$$

1778. The practical method of graduating a valve lever is to attach the valve to the lever and balance both over a knife edge; then, measure the distance from the point of suspension to the center of the pin upon which the lever turns (fulcrum). Calling this distance b and letting $W_3 = W_1 + W_2$ in formula **186,** formula **186** may be written

$$d = \frac{p A a - W_3 b}{W}. \qquad \textbf{(187.)}$$

To show that formula **187** will give the same results as formula **186,** let all conditions remain the same as in the preceding example. Imagine the lever and valve to be balanced on the knife edge and take the center of moments at the balancing point. The weight of the lever per inch of length is $\frac{12}{32} = \frac{3}{8}$ lb. Consequently, applying the rule for moments,

$$\frac{3}{8}(32 - b)\frac{32 - b}{2} - \frac{3}{8} b \times \frac{b}{2} - (b - 3) \times 5 = 0, \text{ or}$$

$$b = 12\tfrac{3}{17}''.$$

Substituting this value of b in formula **187,**

$$d = \frac{75 \times 10 \times 3 - 17 \times 12\frac{3}{17}}{83.1} = 24.58 \text{ in.},$$

the same as before.

1779. Neither of the foregoing methods will give exact results in practice, owing to the slight friction of the moving parts. An exact method for graduating a safety valve lever, or of ascertaining whether a safety valve lever has been graduated correctly or not, is the following, in which an ordinary platform scale may be used:

A 3-inch safety valve has been set to blow off at 100 pounds per square inch, but fails to do so when the steam gauge indicates 100. To ascertain if the lever has been graduated correctly proceed as follows: Remove the valve and cover, and bolt the cover E to a couple of short heavy timbers A and B suspended above the platform of the scale as shown in Fig. 563.

The timbers A and B should be bolted to the floor to prevent them from tipping over. Now, adjust the height

of the valve C so that the lever D will be horizontal. If the lever is too low, slip pieces of sheet iron or other metal under the rollers of the scale until the lever is horizontal; if too high, slip pieces under the cover E. Having gotten the lever into a horizontal position, place the weight W at the 100-lb. notch. The diameter of the valve being 3 in., its area is $3^2 \times .7854 = 7.0686$ sq. in. The total steam pressure necessary to raise the valve when the weight W is at the 100-lb. notch is $7.0686 \times 100 = 706.86$ lb., say 707 lb. If the scale balances when set for 707 lb., the lever has been graduated correctly. If it does not balance at this

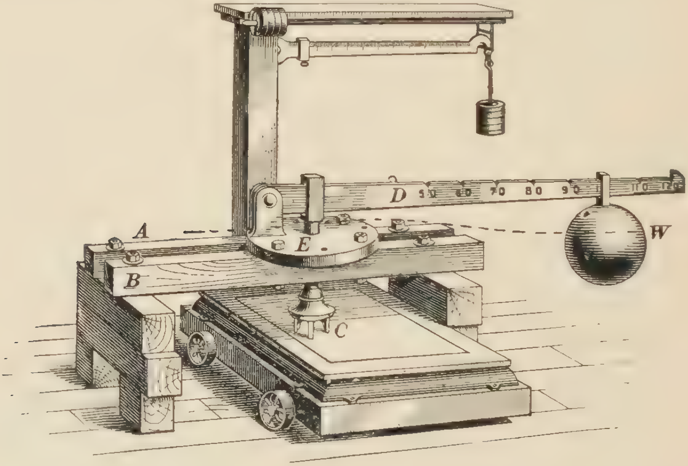


FIG. 563.

point, shift the weight slightly to the right or left until the scale balances; mark this point on the lever, and it will be the correct 100-lb. notch. In the same way, test the graduation of the other points of the scale, for example, the 80-lb. notch. For this point the scale should balance at $7.0686 \times 80 = 565.488$ lb., say 565 lb.

1780. Spring safety valves are mostly used at present, especially on locomotive and marine boilers. The valve is held to its seat by a spring acting either directly on it or on a short lever. The Crosby "pop" safety valve is

shown in Fig. 564. The main valve V is held down on the two circular seats M and N against the steam pressure by the spring S acting on the rod T. The outer seat N is formed on the body A of the valve, while the inner and smaller seat M is formed on the upper edge of a cylindrical chamber B which is connected to the body A by arms containing the passages C, C. The hollow chamber B forms a guide for the valve V. Ordinarily, the steam exerts a pressure on the annular space between M and N; when the valve rises a little the steam rushes over the seat N into the air, and over the seat M into the chamber B, whence it escapes through the channels C, C. The channels are, however, not large enough to allow the steam to escape from the chamber as fast as it enters, and hence the pressure in the chamber rises and acts on the area inside the seat M. This additional pressure throws the valve wide open, and quickly relieves the pressure in the boiler.

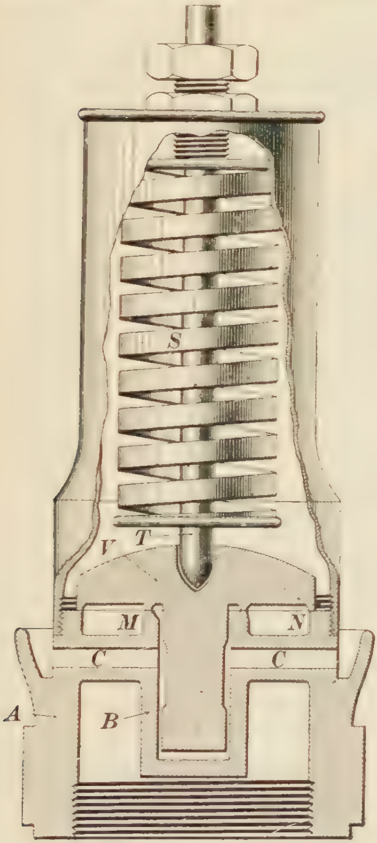


FIG. 564.

1781. The area of a safety valve should be much larger than necessary to discharge the maximum quantity of steam the boiler can make. For finding the area of valves the following formula may be used:

$$a = \frac{.5\,w}{p + 10}, \qquad \textbf{(188.)}$$

where a is the area of the valve opening in sq. in., w the maximum weight of the steam generated in pounds per hour, and p the gauge pressure in pounds per sq. in.

NOTE.—When the area of a safety valve is spoken of, the area of the orifice leading to the bottom of the valve is always meant; in other words, it is the projected area of the surface of the valve in contact with the steam before the valve has opened.

EXAMPLE.—A boiler evaporates 1,800 pounds of water per hour, and works under a pressure of 110 pounds. What should be the area of the valve opening?

SOLUTION.—Using formula **188,** we obtain

$$a = \frac{.5\,w}{p+10} = \frac{.5 \times 1,800}{110+10} = 7\tfrac{1}{2} \text{ sq. in. Ans.}$$

1782. The safety valve should be placed in direct connection with the boiler, so that there is no possible chance of cutting off the communication between them. A stop valve placed between the boiler and safety valve is a very fruitful cause of boiler explosions. Again, the safety valve must be free to act, and to prevent it from corroding fast to its seat, it should be lifted from the seat occasionally. Care must be taken to prevent ignorant persons from raising the blow-off pressure by adding to the weights or increasing the tension of the spring. To this end the weights of lever safety valves are often locked in position by the boiler inspector.

EXAMPLES FOR PRACTICE.

1. A boiler generates 1,400 pounds of steam per hour at a pressure of 85 pounds per sq. in. (*a*) What should be the *diameter* of the safety valve opening? (*b*) Supposing the valve to be of the dead-weight type, what should be the weight?

   Ans. $\begin{cases} (a)\ 3\frac{1}{16} \text{ in.} \\ (b)\ 626 \text{ lb.} \end{cases}$

2. The diameter of the valve opening is $4\frac{5}{8}$ inches, the length of the lever is 35 inches, the distance from fulcrum to valve stem is $3\frac{1}{2}$ inches, and the steam pressure is 80 pounds. The weight of the lever is 16 pounds and of the valve 6 pounds. How far from the fulcrum should a weight of 130 pounds be placed? Ans. 30 in.

3. If the weight in example 2 were hung at the end of the lever, what would be the blow-off pressure? Ans. 92.4 lb.

4. A boiler evaporates 3,500 pounds of water per hour, and generates steam at an average pressure of 95 pounds. What should be the diameter of the safety valve opening? Ans. $4\frac{5}{8}$ in., nearly.

5. Assuming the safety valve of example 4 to be of the lever type, what weight should be placed 40 inches from the fulcrum, the valve being 4 inches from the fulcrum? Neglect the weight of valve and lever. Ans. 159.3 lb.

1783. The **steam gauge** indicates the pressure of the steam contained in the boiler. The most common form is the **Bourdon pressure gauge,** Fig. 565. It consists of a tube *a* of elliptical cross-section, which is filled with water

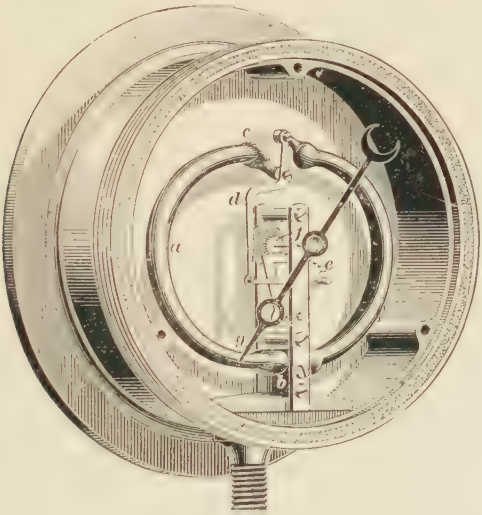


FIG. 565.

and connected at end *b* with a pipe leading to the boiler. The other end *c* is closed and is attached to a link which is in turn connected with a sector *e*; this rack gears with a pinion *f*, to which is attached the index pointer *g*. When the water contained in the elliptical tube is subjected to pressure, the tube tends to take a circular form, and the tube as a whole straightens out, throwing the free end out a distance proportional to the pressure. The movement of the free end is transmitted to the pointer by the link, rack, and

pinion, and the pressure is thus recorded on the graduated dial.

The gauge connections must be so made that the gauge may not be injured by heat. This is accomplished by placing a coil or bend in the gauge pipe as shown in Fig. 566. The coil fills up with comparatively cool water condensed from the steam, which protects the spring from being injured by the heat of the steam.

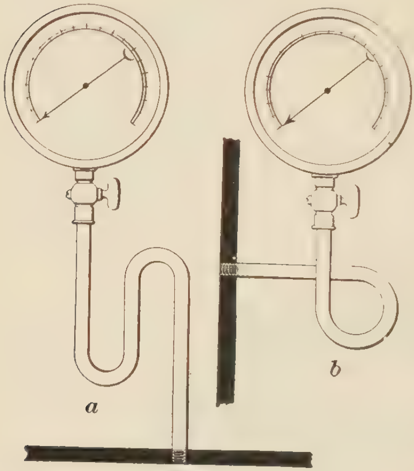


FIG. 566.

1784. Gauge-cocks and **water gauges** indicate the height of the water in the boiler. The cocks, which are three or more in number, are placed on the head, or shell. The lower cock is placed at the lowest level that the water may safely attain, and the upper cock at the highest desirable level. Upon opening a cock above the water-level, steam will issue forth, and upon opening one below the water-level, water will appear. Hence, the level may be easily located by opening the cocks in succession.

1785. The **gauge glass** is a glass tube, the lower end of which communicates with the water space of the boiler, the upper end with the steam space. Hence, the level of the water in the gauge should be the same as in the boiler.

Boilers should be provided with both cocks and gauge glasses. Fig. 567 shows an arrangement of cocks and gauge glass recommended by the Hartford Boiler Insurance Company. *H* is a round cast-iron column, whose inside diameter is about 4″. The upper end communicates with the steam space of the boiler by means of the pipe connection *I*, and the lower end with the water space through the pipe

connection *J*. *K* is a drip pipe for removing the condensed water from the column. The water glass *h* communicates with the column through the connections *L* and *M*. There are three gauge-cocks, *i*, *j*, and *k*. The center line of the lowest one *k* should be located at least 3″ above the level of the tops of the upper row of tubes in the boiler to insure them always being covered with water. The gauge *N* is connected to the pipe *I* by means of the inverted siphon pipe *P*, which answers the same purpose as the bend in Fig. 566.

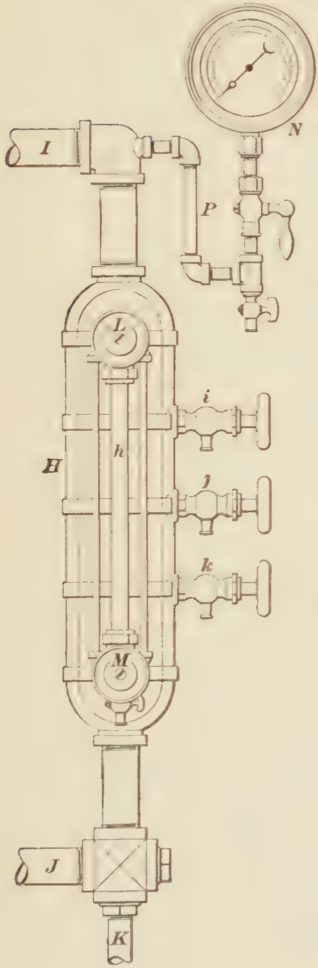


FIG. 567.

1786. Fusible plugs are placed in the crown sheets of furnaces as a safeguard against overheating through shortness of water. The plug consists of an alloy of tin, lead, and bismuth, which melts at a low temperature. So long as the furnace crown is well covered with water, the plug is kept from melting by the comparative coolness of the water, but should the water sink low enough to uncover the top of the plug, it quickly melts and allows the steam and water to rush into the furnace, thus relieving the pressure and extinguishing the fire.

1787. A good form of fusible plug is shown in Fig. 568. The plug *P* is screwed into the crown sheet *Q*, and the fusible cap *R* laid on top of it and kept in place by the

nut U. S is a very thin copper cap placed over the top of R to protect the fusible cap from any chemical action of the water. The top of R extends from $1\frac{1}{2}''$ to $2''$ above the crown sheet, so that when it melts on account of the water being too low, there will still be enough water to cover the crown sheet and prevent its burning. In many places fusible plugs are required by law.

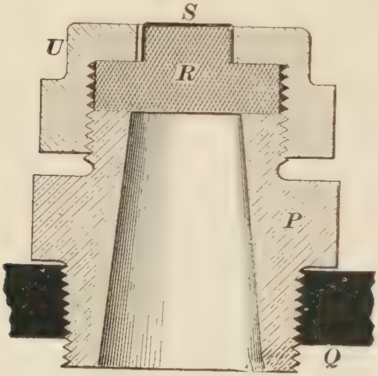


FIG. 5C8.

1788. The **blow-out apparatus** serves the double purpose of emptying the boiler, when necessary, and of discharging the mud and sediment which collects from the feedwater. This latter may be accomplished by partially blowing off the boiler when under steam pressure. The position of the blow-out pipe has been shown in the cuts of most of the boilers described; in ordinary tubular boilers, it is usually led from the bottom of the rear end of the shell through the rear wall; if the boiler is supplied with a mud drum, the blow-out is attached to that.

The blow-out pipe may be closed either by a valve or by a plug tap. The former is the easier to manage, but care must be taken that it does not leak. The valve, though screwed tight, may not be closed properly on account of a chip of incrustation or similar obstruction on the seat; as a result, the water may leak out of the boiler unperceived and an explosion occur. Some good engineers insist that taps only should be used.

The blow-out pipe should be led into a sewer or waste pipe entirely removed from the boiler.

1789. Dome and Dry Pipe.—Domes are placed on cylindrical boilers for the purpose of increasing the steam

space, and also for the purpose of drying the steam, the supposition being that the steam will be dried on account of being further removed from the water. The hole cut in the shell to give communication between the boiler and dome should be made only large enough to allow a man to pass through, since a large hole materially weakens the shell. The edge of the plate around the hole should be reenforced by a wrought-iron ring riveted to it. The flat top of the dome must be stayed by diagonal braces.

In the case of internally fired boilers, the dry pipe has nearly superseded the dome, especially in England. The dry pipe was mentioned in the description of Fig. 496, and a reference to that cut may be made in regard to the manner of supporting it.

1790. The **manhole** gives access to the boiler for the purpose of cleaning or repairing it; it is elliptical in form and large enough to admit a man. The manhole is closed by a cover, or plate *N*, Fig. 569, of cast or wrought iron, the latter being preferable. The plate is held to its seat by one or two yokes *M*, *M* and bolts *Y*, *Y*. The joint between the plate and shell must be made steam-tight by some kind of packing.

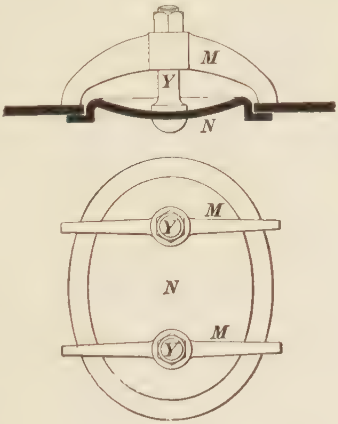


FIG. 569.

The edges of the plate around the manhole should be strengthened by riveting on a wrought-iron ring. The ring may be flanged and used as the seat of the cover; *R*, Fig. 570, shows one form of a ring used in a construction of this kind. The edge of the hole is strengthened by the flanged ring, the inner edge of which is faced to receive the cover, which is also faced. A steam-tight joint is thus formed without the aid of packing.

1791. Mudholes and **handholes** are placed in boilers whose construction does not permit the entrance of a man, as, for example, in vertical boilers. They are also placed in

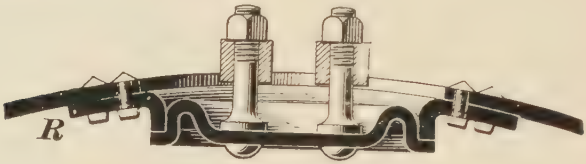


FIG. 579.

other boilers in convenient positions; thus, in a return tubular boiler it is customary to place a handhole in the front head below the tubes. The handhole is a convenient place to rake out sediment and scale, and to admit a hose for the purpose of washing out the boiler.

The handhole and its cover are constructed very much like the manhole and cover; the handhole, being the smaller, requires but one yoke and bolt to close up the cover.

COMBUSTION AND FUELS.

1792. Combustion may be defined as a rapid chemical combination of two or more substances producing heat. The ordinary combustion which takes place in the furnace is the chemical combination of the carbon and hydrogen, of which the fuel is composed, with the oxygen of the air. This chemical combination produces intense heat, which may be used for generating steam or for any other desired purpose.

1793. Carbon and oxygen, or hydrogen and oxygen, will not combine at ordinary temperatures; their temperature must be first raised to a fixed temperature, called the **igniting temperature,** before the attraction between the two is sufficient to cause them to combine. As soon, however, as the combustion is once begun, the temperature is kept up to the igniting point, or above it, by the combustion itself, and, as long as the oxygen and carbon or hydrogen are supplied the combustion will continue.

1794. Products of Combustion.—The combination of oxygen with the carbon and hydrogen of the fuel produces gaseous compounds, which pass away through the flues and chimney. Carbon and oxygen form two compounds, *carbon monoxide*, usually called **carbonic oxide,** and *carbon dioxide*, usually called **carbonic acid.** Hydrogen and oxygen form a single compound, steam (or water). Carbonic acid is the product of *complete combustion* of carbon; each pound of carbon unites with $2\frac{2}{3}$ pounds of oxygen and forms $3\frac{2}{3}$ pounds of carbonic acid. Carbonic oxide is the product of *incomplete combustion;* the pound of carbon in this case unites with $1\frac{1}{3}$ pounds of oxygen to form $2\frac{1}{3}$ pounds of carbonic oxide. The formation of carbonic oxide is an incomplete combustion, because the carbon does not combine with as large a quantity of oxygen as it is possible for it to do. The $2\frac{1}{3}$ pounds of carbonic oxide may itself combine with another $1\frac{1}{3}$ pounds of oxygen, thus completing the combustion and forming the $3\frac{2}{3}$ pounds of carbonic acid. One pound of hydrogen requires 8 pounds of oxygen for complete combustion, the resulting product being 9 pounds of steam or water.

1795. Sulphur appears as a combustible constituent in some fuels, but in such small quantities that its presence may be neglected. We may safely assume that all fuels used for engineering purposes are composed of carbon and hydrogen, and, in some cases, oxygen.

1796. The oxygen required for combustion is taken from the air. Air is composed of 23 parts (by weight) of oxygen and 77 parts of nitrogen. More exactly, the per cent. of oxygen is 23.185 and of the nitrogen 76.815, but for practical purposes, and for the solution of the following problems, the values 23% and 77% may be used. The nitrogen takes no part in the combustion, and simply passes off through the chimney, carrying a certain quantity of heat along with it.

The process of combustion in air may be shown by the following diagrams:

COMBUSTION OF CARBON—COMPLETE.

Elements.	Process.	Products.
1 lb. carbon....	1 lb. carbon	.... 3.67 lb. car. acid
11.61 lb. air.....	2.67 lb. oxygen	
	8.94 lb. nitrogen......	8.94 lb. nitrogen
12.61 lb.	12.61 lb.	12.61 lb.

COMBUSTION OF HYDROGEN.

1 lb. hydrogen	1 lb. hydrogen	.... 9 lb. water
34.8 lb. air......	8 lb. oxygen	
	26.8 lb. nitrogen......	26.8 lb. nitrogen
35.8	35.8	35.8

The weight of air required for complete combustion is, of course, $\frac{100}{23} = 4.35$ times the weight of oxygen required for the combustion, since air is 23 per cent. oxygen. Thus, as 1 pound of hydrogen requires 8 pounds of oxygen for complete combustion, it requires $8 \times 4.35 = 34.8$ pounds of air for the same purpose.

1797. The **minimum quantity of air** required for the complete combustion of any given fuel may readily be found when its chemical composition is known. It has been shown that a pound of carbon requires 11.6 pounds of air, and a pound of hydrogen requires 34.8 pounds of air for complete combustion. Letting C represent the per cent. of carbon and H the per cent. of hydrogen contained in the fuel, the minimum quantity of air required for the complete combustion of a pound of fuel must be $11.6\,C + 34.8\,H$. If the fuel contains oxygen, the oxygen will unite with $\frac{1}{8}$ its weight of hydrogen to form water, and the weight of air (W) required will be

$$W = 11.6\,C + 34.8\left(H - \frac{O}{8}\right), \qquad \textbf{(189.)}$$

where O represents the percentage of oxygen in the fuel.

EXAMPLE.—A certain kind of coal has the following chemical composition:

Carbon	.80
Hydrogen	.08
Oxygen	.12
	1.00

Find the minimum quantity of air required for complete combustion of one pound of coal.

SOLUTION.—Using formula **189**,

$$W = 11.6\,C + 34.8\left(H - \frac{O}{8}\right) = 11.6 \times .8 + 34.8\left(.08 - \frac{.12}{8}\right) = 11.54 \text{ lb.}$$

Ans.

1798. The minimum *volume* of air in cubic feet may be found by multiplying the result obtained from formula **189** by 13.14, which is the number of cubic feet occupied by a pound of air at 62° F.

1799. Owing to the difficulty of perfectly mixing the air with the thick bed of burning fuel, complete combustion cannot be attained in practice by using the theoretical quantity of air given by the formula.

Furnaces with ordinary chimney draft require about double, and furnaces with forced draft about $1\frac{1}{2}$ times the theoretical quantity of air.

1800. Heat of Combustion.—It has been determined by direct experiment that a pound of carbon, burned to carbonic acid gas, gives out 14,500 British thermal units of heat; burned to carbonic oxide, it gives out only 4,400 B. T. U. The complete combustion of a pound of hydrogen gives out 62,032 B. T. U. The heat of combustion of a given fuel may be determined approximately by the following formula:

$$h = 14{,}500\,C + 62{,}032\left(H - \frac{O}{8}\right), \qquad \textbf{(190.)}$$

where h is the heat of combustion in heat units and C, H, and O have the same meaning as in formula **189.**

EXAMPLE.—The composition of a variety of coal is as follows: Carbon, 82%; hydrogen, 4%; oxygen, 6%; other substances, 8%; total, 100%. Find the approximate amount of heat developed by the combustion of one pound of the coal.

SOLUTION.—Using formula **190**,

$$h = 14,500\,C + 62,032\left(H - \frac{O}{8}\right) = 14,500 \times .82 + 62,032\left(.04 - \frac{.06}{8}\right) =$$

13,906 B. T. U. Ans.

1801. Formula **190** gives only approximate results, and where the heating power of a fuel needs to be accurately known, as, for example, in evaporative tests of steam boilers, it is better to determine the heat of combustion by direct calorimetric experiment.

The total heat required to raise a pound of water at 62° F. to 212° F. and completely evaporate it at the latter temperature is, from the steam table, 1,116.6 B. T. U., and the heat required to evaporate a pound of water at 212° is 966.1 B. T. U. Consequently, a pound of carbon completely burned to carbon dioxide should raise $\frac{14,500}{1,116.6} = 13$ pounds of water, nearly, from 62° to 212° and evaporate it at the latter point, or it should vaporize $\frac{14,500}{966.1} = 15$ pounds of water from and at 212° F. In a similar manner the evaporative power of any fuel may be obtained by dividing its heat of combustion by 1,116.6 for an evaporation from 62° or by 966.1 for an evaporation from 212°.

In actual practice the above theoretical values of the evaporative powers of fuels can not be attained on account of the loss of heat by radiation and other causes. The best boilers can evaporate about 11 to 12½ pounds of water from and at 212° per pound of combustible.

1802. The **temperature of the fire** depends upon the total heat of combustion and upon the products of combustion. The specific heats of the products of combustion are as follows (see Table 21, Art. **1134**):

Air	.23751	Carbonic acid	.217
Oxygen	.21751	Nitrogen	.2438
Steam	.4805	Carbonic oxide	.2479

Suppose we take the case of the complete combustion of carbon. It was found that per pound of carbon there were given off 3.67 pounds of carbonic acid and 8.94 pounds of

nitrogen; and the heat of combustion was found to be 14,500 B. T. U. per pound of carbon.

The heat of combustion is given up to the products of combustion, the carbonic acid and nitrogen. The heat required to raise these products one degree in temperature is $3.67 \times .217 + 8.94 \times .2438 = 2.976$ B. T. U. Hence, the total rise in temperature above atmosphere is $14,500 \div 2.976 = 4,872°$ F.

To take a more complicated case, suppose the fuel is coal containing the elements in the following proportions: Carbon .84, hydrogen .06, and oxygen .10. Required to find the temperature of the furnace, supposing that double the theoretical quantity of air is mixed with the fuel.

Making allowance for the water due to combination of H and O in the fuel, we have for the free hydrogen,

$$H - \frac{O}{8} = .06 - \frac{.10}{8} = .0475,$$

and the composition of the coal may be written, carbon .84, hydrogen .0475, and water .1125, since $.10 + .06 = .16$ and $.16 - .0475 = .1125$. The heat of combustion per pound of coal is, therefore, by formula **190,** $.84 \times 14,500 + .0475 \times 62,032 = 15,126.52$ B. T. U.

The minimum air required is, by formula **189,** $11.6 \times .84 + 34.8 \times .0475 = 11.4$ pounds, nearly; the air furnished is, therefore, $11.4 \times 2 = 22.8$ pounds.

The products of the combustion are

Carbonic acid..................	$.84 \times 3.67 =$	3.08 lb.
Water (steam)...........	$.1125 + .0475 \times 9 =$	.54 lb.
Nitrogen	$11.4 \times .77 =$	8.78 lb.
Free air..................................		11.4 lb.
		23.8 lb.

The extra 11.4 pounds of air simply pass up the chimney with the other products of combustion. The object of supplying it was to insure perfect combustion.

From the total heat of combustion must be subtracted the heat required to change the water into steam, that is, $.54 \times 966.1$ B. T. U. The resulting temperature is

$$t = \frac{15{,}126.5 - .54 \times 966.1}{3.08 \times .217 + .54 \times .4805 + 8.78 \times .2438 + 11.4 \times .2375}$$
$= 2528.5°$ F. above the temperature of the atmosphere.

If the theoretical quantity of air had been supplied instead of double that amount, the temperature would have been about 4800° F.; hence, it is apparent that air in excess of minimum amount required for combustion dilutes the products of combustion and lowers the temperature of the furnace. It is very important that just the proper amount of air be supplied to insure complete combustion; a greater quantity leads to loss of heat by diluting the products of combustion, while, on the other hand, an insufficient quantity leads to loss through incomplete combustion.

1803. The **rate of combustion** of the fuel in the furnace is usually stated in pounds per hour burned on each square foot of grate.

For coal the usual rates of combustion are as follows:

CHIMNEY DRAFT.

	Pounds per sq ft. per hour.
1. Slowest rate of combustion in Cornish boilers	4 to 6
2. Ordinary rate of combustion in Cornish boilers	10 to 15
3. Ordinary rates in factory boilers	12 to 18
4. Ordinary rates in marine boilers	15 to 25
5. Quickest rates of complete combustion of anthracite coal, the supply of air coming from the grate only	15 to 20
6. Quickest rates of complete combustion of bituminous coal, with airholes above the fuel $\frac{1}{36}$ area of grate	20 to 25

FORCED DRAFT.

7. Locomotives	40 to 100
8. Torpedo boats	60 to 125

1804. The **fuels** used in the generation of steam are chiefly coal, coke, wood, the mineral oils, such as petroleum, and natural gas. Other fuels are sometimes used under

exceptional conditions; such as straw, bagasse (refuse from sugar cane), dried tan bark, and peat.

All of these fuels are composed either of carbon alone or carbon in combination with hydrogen, oxygen, and non-combustible substances.

1805. There are five leading varieties of *coal:*

1. Anthracite, or hard coal, consisting almost entirely of pure carbon.
2. Dry bituminous coal, containing 70 to 80 per cent. carbon.
3. Bituminous coking coal, containing 50 to 60 per cent. carbon.
4. Cannel coal, containing 70 to 85 per cent. carbon.
5. Lignite, or brown coal, containing 55 to 75 per cent. carbon.

Anthracite coal is difficult to burn, and requires a strong draft, high temperature, and much attention. It burns without flame or smoke, which gives it a peculiar value for some purposes.

The bituminous coals burn freely and rapidly, and give off flame and smoke. The best bituminous coals have a higher heating value than anthracite, and are more highly esteemed as steam coals.

Lignite is a sort of incomplete coal, and is not a very valuable fuel.

Coke is made from bituminous coal by driving off its volatile constituents. It is used chiefly for metallurgical purposes, though it is a valuable fuel for steam generation.

Wood is much used for steam purposes in localities where it is abundant. The chemical composition of ordinary seasoned firewood is as follows:

Carbon	.375
Hydrogen	.045
Oxygen	.3075
Nitrogen	.0075
Ash	.015
Water	.25
	1.0000.

The effective heating value of all kinds of wood is about the same, and is usually estimated at 0.4 the heating value of the same weight of coal.

Slack, or screenings from coal, when burned upon a properly constructed furnace with a blast, give nearly as good heating results as coal.

Petroleum is coming into quite common use as a fuel for boilers, and offers many advantages, among which are the ease of lighting and controlling the fire, the uniformity of the combustion, and the economy of labor. The objections to the use of petroleum are: Danger of explosion, loss of fuel by evaporation, and high price of petroleum.

The average chemical composition of petroleum is as follows:

Carbon	.847
Hydrogen	.131
Oxygen	.022
	1.000.

A pound of petroleum completely burned generates about 20,500 B. T. U.

Natural gas is abundant in parts of Ohio and Pennsylvania, and is used as a boiler fuel. It is worth from 2 to $2\frac{1}{2}$ times the same weight of coal, or about 30,000 cubic feet are equal to a ton of coal.

Attempts have been made without great success to use artificial gas of various kinds for fuel purposes. Their chief disadvantage is high cost.

1806. The **transfer of heat** from the furnace to the water of the boiler is accomplished by radiation, conduction, and convection. It is estimated that when the fire is burning brightly about one-half of the heat received from the furnace by the boiler is radiated. The transfer of heat through the water is due to convection, since liquids are poor conductors of heat. The particles of water next to the shell become heated and immediately rise into the main body of water, giving place to fresh particles of cold water. The rapidity with which heat will be absorbed by convec-

tion, therefore, depends upon the effectiveness of the water circulation in the boiler, and upon the extent and conductivity of the heating surfaces. The transference of heat through the shell and furnace plates takes place by conduction. It has been experimentally shown that the quality or thickness of the material has little influence, thick iron tubes working practically as well as thin brass ones. Very thick plates are, however, liable to be injured by burning when exposed to direct action of the fire, and hence in some cases the thickness of furnace plates is limited by law.

1807. Water circulation is essential to the efficient operation of a boiler. It has just been shown that the rapidity of the transfer of heat by convection depends upon the rapidity of circulation; besides this, the circulation is useful in preventing, to a certain extent, the deposits of

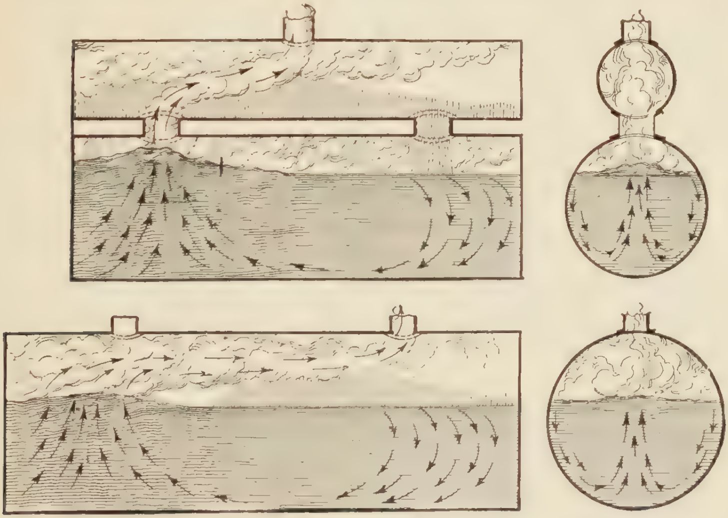


FIG. 571.

sediment which accumulate from the feed-water. Again, a rapid circulation keeps the parts of the boiler at a uniform temperature, and is a safeguard against overheating.

Fig. 571 shows the circulation of water in a plain cylindrical boiler. The heated currents rise from the hottest part

of the shell directly over the furnace, and carry the bubbles of steam to the surface. The cooler water rushes in to take their places over the furnace, and thus the circulation is maintained. As shown in the figure, there are two currents; one carrying the cold water from rear to front, and the other carrying it down the outside of the shell and up through the center. It will also be noticed that the circulation is in a direction contrary to that of the furnace gases.

Since in all cylindrical shells the water is contained in a solid mass, broken only by flues or tubes, the circulation is more or less interfered with by opposing currents. The circulation is more rapid and effective if the water is constrained to follow a particular path. This is one of the

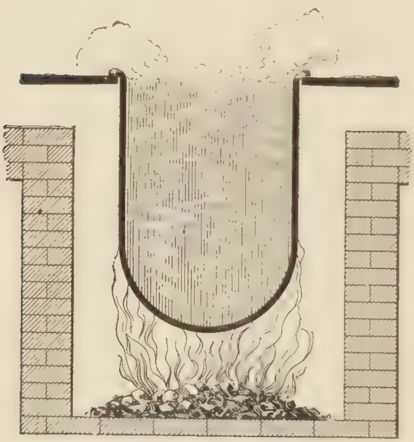


FIG. 572.

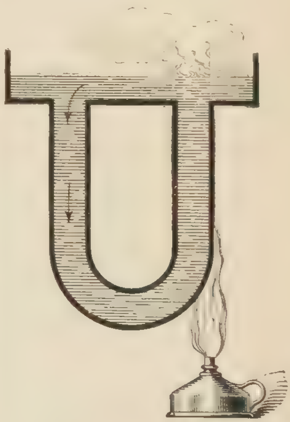


FIG. 573.

strong points of the water-tube boiler. The water *must* pass in one direction through a series of tubes; hence, the circulation is strong and uninterrupted. The difference between the cylindrical and water-tube boilers in this respect may be illustrated as follows: The cylindrical boiler, with its contained mass of water, may be compared to an ordinary kettle in the process of boiling. (See Fig. 572.) The water rises rapidly around the outer edges, and flows downwards in the center. If, however, the fire is quickened, the upward and downward currents interfere and the kettle

boils over. The water-tube boiler is identical in principle with a **U** tube depending from a vessel of water (Fig. 573) with the heat applied to one leg. The circulation is set up immediately and proceeds quietly, no matter how fierce the fire may be.

1808. Loss of Heat.—A portion of the heat generated in the furnace is usefully expended in evaporating water. There is, however, more or less loss of heat due to the following causes:

1. A certain amount of heat is carried up the chimney by the waste gases, the temperature of which is between 500° and 700°.

2. In some cases heat is lost by incomplete combustion.

3. Heat is required to vaporize the water formed by the combustion of hydrogen.

4. The escape of free carbon in the form of smoke is a loss.

5. Heat is lost by radiation.

The loss due to the first of the above causes can not be avoided where chimney draft is used, since the temperature of the ascending gases must be 300° to 600° F. to insure a good draft. The loss may, however, be aggravated by using an excess of air.

The loss due to the second cause may be avoided by using a sufficient quantity of air to insure complete combustion.

The loss of the heat required to vaporize the water is unavoidable; it is, of course, much greater with wet fuels.

The formation of smoke is a very common and very fruitful source of waste. Bituminous coals which are rich in hydro-carbon gases produce the most smoke. The volatile hydro-carbons are driven off by the high temperature, and come in contact with the air; the oxygen of the air unites with the more inflammable hydrogen, leaving the carbon in a finely divided condition; upon being cooled down, the fine particles of carbon appear as smoke.

Smoke may be avoided by bringing a supply of fresh air in contact with the carbon while it is red hot in the flame;

that is, before it cools down into smoke. After being formed smoke is hard to burn, and the skill of the fireman should be exerted in preventing its formation. A successful method of smoke prevention, devised by D. K. Clark, consists in forcing air into the furnace *above* the fuel by means of steam jets.

The loss of heat by radiation may be made small by covering exposed parts of the boiler with non-conducting materials. The radiation from an internally fired boiler is less than from one externally fired, the excess of the latter over the former being due to the brickwork furnace walls.

EXAMPLES FOR PRACTICE.

1. A certain coal has the following chemical composition:

Carbon	90.4 per cent.
Hydrogen	3.3 per cent.
Oxygen	3.0 per cent.
Other substances	3.3 per cent.
	100.0

Calculate the heat of combustion of one pound.

Ans. 14,922.4 B. T. U.

2. The chemical composition of some marsh gas is,

Carbon	85.7 per cent.
Hydrogen	14.3 per cent.

(*a*) Find the heat of combustion of one pound.

(*b*) Find the weight of air per pound necessary for complete combustion.

Ans. { (*a*) 21,297 B. T. U. (*b*) 14.92 lb.

3. The chemical composition of a certain kind of peat is as follows:

Carbon	60 per cent.
Hydrogen	6 per cent.
Oxygen	31 per cent.
Ash	3 per cent.

(*a*) Find the heat of combustion of one pound of peat.

(*b*) Find the weight of air per pound of peat required for complete combustion.

Ans. { (*a*) 10,018.2 B. T. U. (*b*) 7.7 lb.

THE FURNACE—FURNACE FITTINGS.

1809. The proper combustion of the fuel depends to a great extent upon the proper design of the furnace and its fittings. The furnace, as has already been shown, comprises

the grate, ash pit, bridge, opening for introduction of fuel, doors, and dampers.

1810. The **grate,** which supports the fuel, is usually made of cast-iron bars (*A*, Fig. 574) placed side by side, and supported by wrought-iron bearers. The lugs cast on each bar determine the size of the air space between them. For anthracite coals the air space is $\frac{3}{8}$ to $\frac{1}{2}$ inch wide, while for coals that cake much the width of space may be $\frac{3}{4}$ inch. The bars are about $\frac{3}{4}$ inch wide at the top and taper towards the bottom. For long furnaces the bars are generally made in two lengths of about 3 feet each, with a bearer

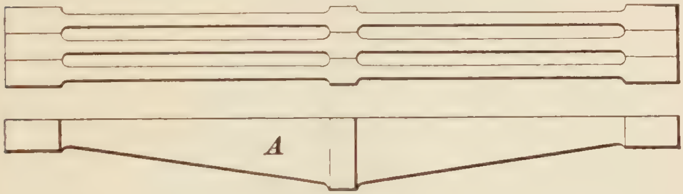


Fig. 574.

in the middle of the grate. Long grates are generally set with a slope towards the bridge to facilitate the firing. In stationary fire-box boilers, and vertical boilers, it is preferable to support the grate independent of the shell of the furnace.

1811. Shaking grates are to some extent taking the place of the ordinary grate. By their use the fire may be cleaned with little labor and without opening the fire door.

The McClave shaking grate is shown in Fig. 575. The grate bars *e*, *e*, etc., are supported by trunnions *c*, *c* which fit into a bearing bar on both sides of the grate. A portion of the bearing bar has been removed in order to show the grate bars properly. By working the lever *l* backwards and forwards, the rod *r* which is connected to the lever *l* by the stub lever *m*, and also with the grate bars at *n*, *n*, etc., transmits the motion of the lever to the grate bars and causes them to rotate to and fro on the trunnions.

The grate operates on the pocket principle; when the grate bars are thrown wide open, they form a series of pockets

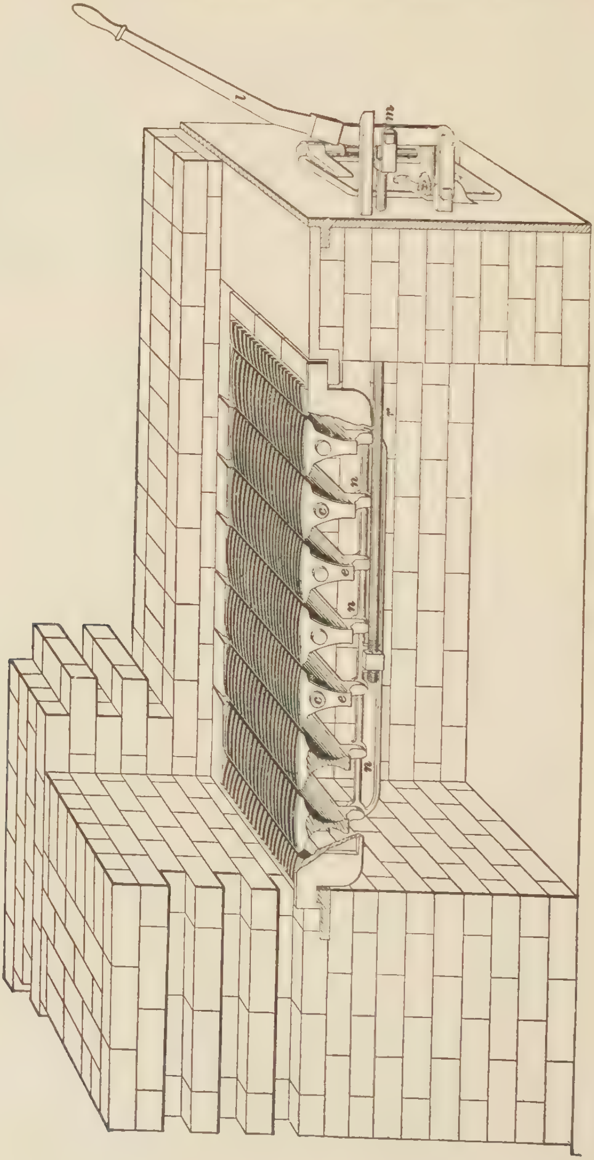


FIG. 575.

which receive the ashes and clinkers; when the bars are thrown back to their upright position the ashes are deposited in the ash pit, and thus a quantity of ashes is removed from the bottom of the fire at each sweep of the lever.

This grate has been recently improved so as to allow one-half of the grate to be shaken without disturbing the other half. This is an advantage, inasmuch as the front half of the fire may be all right, while the back half may need shaking, and *vice versa*.

This method of cleaning is particularly applicable to anthracite coal fires, since the main body of the fuel is left undisturbed. The grate may also be used for shaking and breaking up the solid bed of fuel, as is necessary in fires of caking bituminous coals. For this purpose the lever is vibrated rapidly back and forth through a small arc. The ashes are shaken through the grate, and the mass of fuel is broken up. This grate is extensively used for burning the smaller sizes of anthracite, such as the pea and buckwheat sizes, and culm.

1812. The Argand steam blower (Fig. 576) is used in

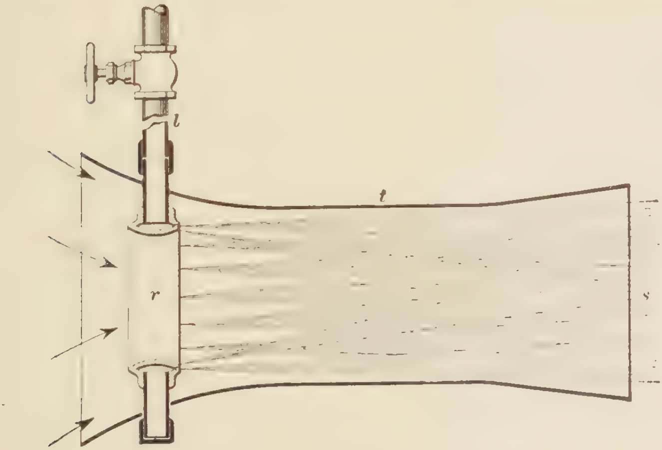


FIG. 576.

connection with the McClave grate to furnish a forced draft. The blower consists of a long air tube *t* discharging from

the end *s* into the space below the grate. In the other end of the tube is placed a ring-shaped tube *r* perforated on the right with small holes around the face. Steam from the boiler is led into the ring by the pipe *l* and escapes in jets through the perforations, carrying air along with it through the opening *r* into the ash pit. This method of producing a draft by means of an air blast below the grate is particularly valuable in burning the small sizes of anthracite coal.

1813. The **bridge** is a low wall at the back end of the grate; it forms the rear end of the furnace. The bridge is usually built of firebrick, though in some cases it is made of wrought iron, with an interior water space communicating with the inside of the boiler. It is the duty of the bridge to bring the flame in close contact with the heating surface of the boiler. The passage between the bridge and boiler shell should not be too small; its area may be approximately $\frac{1}{6}$ the area of the grate. Likewise, the space between the grate and shell should be ample for complete combustion. Thurston advises that the distance between grate and boiler shell should be $\frac{1}{2}$ the diameter of the shell.

1814. Mechanical stokers are used in many boiler plants in preference to firing by hand. The fuel is fed into the hopper and rests upon a *feed plate* which is given a back and forth motion by a pusher. The fuel is thus gradually pushed forward on to another plate called a **dead plate,** and thence on to the grate. The shaft of a disk crank is connected with some source of power and rotates from 7 to 10 times a minute. The motion of this crank is transmitted through the link to the pusher, and likewise to the connecting-rod; the latter, connecting with the grate bars, gives them a continuous rocking motion, which gradually works the coal down the grate, and finally lands the ash and cinders upon the dumping grate, from which they are dumped into the ash pit. The rate of feeding may be adjusted by the hand wheel, and the rock of the grate bars may be adjusted by the nuts on the rod. Mechanical stokers are particularly

valuable for furnaces burning the small sizes of coal, culm, slack, or sawdust.

FIRING.

1815. The thickness of the bed of fuel depends upon the nature of the fuel and the force of the draft; a strong draft demands a thick fire, a light draft a thin one. The larger sizes of coal may be burned in deeper beds than the smaller, and in general, hard coals can be used in beds of greater depth than can the soft coals. With ordinary chimney draft, the depth of the fire varies from 3 to 14 inches, while with forced draft, as in locomotive or marine boilers, the depth may be greatly increased.

1816. There are three styles of hand firing ordinarily practised, **spreading, alternate,** and **coking firing.** Where spreading firing is employed the coal is spread evenly over the whole surface of the fire, commencing at the bridge and working towards the door. In alternate firing the coal is thrown alternately on each side of the furnace; at one firing, one side of the grate is spread with coal, and at the next firing, the other side receives the charge. This method is preferable to the spreading system in that the whole of the furnace is not cooled off at once by the fresh fuel. The coking system of firing is used with bituminous coals containing a large proportion of hydro-carbons. The coal is first piled on the dead plate near the door and there allowed to coke. The hydro-carbon gases are driven off and burned at the front of the furnace, and then the coal is pushed backwards towards the bridge, and another charge of fresh coal placed on the dead plate.

The smaller sizes of coal can not be burned economically without almost continuous firing with small charges. It is best to use a mechanical stoker in this case.

1817. The points to be observed in successful firing are as follows: Find the best depth for the fire and keep it at that thickness. Fire often enough to prevent the fire from

burning into holes, or in irregular thickness. Never allow the back of the grate to become bare. Fire quickly to avoid as much as possible the rush of cold air into the furnace. The ash pit should be kept clean to avoid checking the draft or burning out the grates. If the ash pit is shallow it is often filled with water to a depth of about two inches. There are two methods of cleaning a fire by hand. In the first method, the live coal is pushed to the left side of the furnace, the right side of the grate is cleaned, and all the live coal is put on this side. The left side of the grate is now cleaned, the live coal is spread evenly over the grate, and a light charge of fresh coal put on. In the second method, the live coal is pushed back against the bridge, the front half of the grate is cleaned, the live coal is pulled to the front, and the clinkers and cinders on the back of the grate are pulled forwards over the top of the live coal. The live coals are then spread and lightly covered with fresh fuel. The damper in the chimney should be partially closed while cleaning the fire to prevent the rush of cold air cooling the hot parts of the boiler. For the same reason, and to prevent the loss of steam pressure as well, the cleaning should be done as quickly as possible.

1818. Liquid fuels are generally sprayed into the furnace by jets of steam or compressed air. The chief care of the fireman is to guard against an explosion arising from the vapors of the fuel, and to regulate the supply of air so as to obtain the most economical combustion.

FORCED AND NATURAL DRAFT.

1819. The necessary quantity of air is supplied to a furnace by a draft—either artificial or natural.

Forced draft, or artificial draft, is produced mechanically by fans or blowers; natural draft is produced by a chimney, the impelling force being the difference between the weight of the hot gases in the chimney and the weight of an equal column of cold air.

1820. Forced draft is extensively used in marine work. The draft is practically applied in one of three ways:

1. The air is forced by a fan into an air-tight fireroom.
2. The air is forced by the fan into air-tight ash pits.
3. The fan is placed in the uptake, and by driving out the waste gases creates a partial vacuum, which draws in the air through the grates.

The pressure of the draft is measured in inches of water; thus, in the above cases the air is forced into the fireroom or ash pit under a pressure of 1 to 3 inches of water; an inch of water corresponding to a pressure of .036 pound per square inch, or about 5.2 pounds per square foot.

1821. The horsepower necessary to furnish a forced draft may be calculated by the following formula:

Let p = pressure of draft in pounds per square foot;
W = weight of fuel burned per minute;
V = volume of air in cubic feet per pound of fuel;
y = efficiency of draft apparatus.

$$\text{Then, H. P.} = \frac{p\,W\,V}{33{,}000\,y}. \qquad \textbf{(191.)}$$

EXAMPLE.—What horsepower is required to supply air at a pressure of $2\frac{1}{2}$ inches of water to a total grate area of 120 sq. ft., burning 50 pounds of coal per sq. ft. per hr., and requiring 220 cu. ft. of air per pound of coal? Assume the efficiency to be 60 per cent.

SOLUTION.—By formula **191**,

$$\text{H. P.} = \frac{p\,W\,V}{33{,}000\,y} = \frac{2\frac{1}{2} \times 5.2 \times \frac{50}{60} \times 120 \times 220}{33{,}000 \times .60} = 14.44 \text{ horsepower. Ans.}$$

1822. The third method of producing a forced draft mentioned above is at present being applied successfully to stationary boiler plants. It is particularly applicable where economizers are used, since the draft depends in no way upon

the temperature of the furnace gases. The use of economizers with ordinary chimney draft has in some cases resulted in a decrease of economy, because the lowering of the temperature of the waste gases weakened the draft so much as to overbalance the gain of heating the feed-water. By using a forced draft, however, the temperature of the gases may be reduced to the lowest possible point without affecting the draft pressure. When so used, in connection with an economizer, the gases of combustion pass from the boilers into the main flue which communicates with the fan. When the fan is in operation, the gases are sucked through the flue and forced out of the chimney, which rises a short distance above the building. The economizer is placed in the main flue and extracts the heat from the gases to raise the temperature of the feed-water. It consists of a series of vertical pipes connected top and bottom, and so arranged that the feed-water circulates through them in a direction opposite to the course of the gases. An average of several tests of this system showed that the gases were reduced in temperature from 525° to 270° F. and that the temperature of the feed-water was raised from 150° to 295° F., representing a gain in fuel of about 14 per cent.

1823. Forced draft may also be produced by blowers, actuated by jets of steam, as in the Argand blower already referred to. The forced draft of locomotives is produced by forcing the exhaust steam through the smokestack, as described in Art. **1077.**

1824. The **chimney** serves the double purpose of creating a draft and carrying away obnoxious gases. The production of the draft depends upon the fact that the furnace gases (the products of combustion) passing up the chimney have a high temperature, and are consequently lighter than an equal volume of outside air at ordinary temperature; that is, the pressure within the chimney is slightly less than the pressure of the outside air. Consequently, the air will flow from the place of higher pressure to the place

of lower pressure—that is, into the chimney through the furnace.

Suppose, for example, the average temperature of the gases in a chimney 150 feet high is 500° F. A pound of burned gases at 62° F. has a volume of 12.5 cu. ft.; its volume at 500° is, then, $\frac{12.5 \times (500 + 460)}{62 + 460} = 23$ cubic feet. Therefore, a column of burned gases 1 foot square and 150 feet long would weigh $\frac{150}{23} = 6.52$ pounds. A similar column of air at 62° F. would weigh $\frac{150}{13.14} = 11.42$ pounds, nearly. Hence, the pressure of the draft is $11.42 - 6.52 = 4.9$ pounds per sq. ft. = .947 in. of water. It is evident that the pressure of the draft depends upon the temperature of the furnace gases and the height of the chimney. The higher the chimney, the lower may be the temperature of the gases to produce the same draft, and the greater will be the economy of the furnace. In general, chimneys are not built much less than 100 feet in height.

1825. The relation between the height of the chimney and the pressure of the draft in inches of water is given by the following formula:

$$p = H\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right), \qquad \textbf{(192.)}$$

where p is the draft in inches of water, H the height of chimney in feet, and T_a and T_c the absolute temperature of the outside air and of the chimney gases, respectively.

Example.—What draft pressure will be produced by a chimney 120 feet high, the temperature of the chimney gases being 600° F.; of the external air 60° F.?

Solution.—By formula **192**,

$$p = H\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right) = 120\left(\frac{7.6}{460 + 60} - \frac{7.9}{460 + 600}\right) = .85 \text{ in. Ans.}$$

The draft pressures ordinarily produced by chimneys vary from 0 to 2 inches of water. Wood requires least draft, and

the small sizes of anthracite coal the greatest draft. To successfully burn anthracite, slack, or culm, a draft of $1\frac{1}{4}$ inches is necessary.

1826. To find the height of chimney to give a specified draft pressure, formula **192** may be transformed.

$$H = \frac{p}{\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right)}. \qquad \textbf{(193.)}$$

Example.—Required, the height of the chimney to produce a draft of $1\frac{1}{8}$ inches of water, the temperature of the gases and of the external air being, respectively, 550° and 62°.

Solution.—By formula **193**,

$$H = \frac{p}{\left(\frac{7.6}{T_a} - \frac{7.9}{T_c}\right)} = \frac{1.125}{\frac{7.6}{522} - \frac{7.9}{1010}} = 167 \text{ feet. Ans.}$$

1827. The height of the chimney being decided upon, its cross-sectional area must be designed to easily carry off the products of combustion. The following formulas for finding the dimensions of chimneys are in common use :

Let H = height of chimney in feet;
$H.\ P.$ = horsepower of boiler or boilers;
A = actual area of chimney in square feet;
E = effective area of chimney in square feet;
S = side of square chimney in inches;
d = diameter of round chimney in inches.

Then, $$E = \frac{.3\ H.\ P.}{\sqrt{H}} = A - .6\sqrt{A}. \qquad \textbf{(194.)}$$

$$H.\ P. = 3.33\ E\sqrt{H}. \qquad \textbf{(195.)}$$

$$S = 12\sqrt{E} + 4. \qquad \textbf{(196.)}$$

$$d = 13.54\sqrt{E} + 4. \qquad \textbf{(197.)}$$

Table 38 has been computed from the above formulas

TABLE 38.
SIZE OF CHIMNEYS AND HORSEPOWER OF BOILERS.

Height of Chimney in Feet.											Effective Area in Sq. Ft.	Actual Area in Sq. Ft.	Side of Square. Inches.	Diameter in Inches.
50	60	70	80	90	100	110	125	150	175	200				
Commercial Horsepower.														
23	25	27									0.97	1.77	16	18
35	38	41									1.47	2.41	19	21
49	54	58	62								2.08	3.14	22	24
65	72	78	83								2.78	3.98	24	27
84	92	100	107	113							3.58	4.91	27	30
	115	125	133	141							4.47	5.94	30	33
	141	152	163	173	182						5.47	7.07	32	36
		183	196	208	219						6.57	8.30	35	39
		216	231	245	258	271					7.76	9.62	38	42
			311	330	348	365	389				10.44	12.57	43	48
			402	427	449	472	503	551			13.51	15.90	48	54
			505	539	565	593	632	692	748		16.98	19.64	54	60
				658	694	728	776	849	918	981	20.83	23.76	59	66
				792	835	876	934	1023	1105	1181	25.08	28.27	64	72
					995	1038	1107	1212	1310	1400	29.73	33.18	70	78
					1163	1214	1294	1418	1531	1637	34.76	38.48	75	84
					1344	1415	1496	1639	1770	1893	40.19	44.18	80	90
					1537	1616	1720	1876	2027	2167	46.01	50.27	86	96

EXAMPLE.—What should be the diameter of a chimney 100 feet high, which furnishes draft for a 600 horsepower boiler?

SOLUTION.—By formula **194**,

$$E = \frac{.3\ H.\ P.}{\sqrt{H}} = \frac{.3 \times 600}{\sqrt{100}} = 18.$$

Now, using formula **197**,

$$d = 13.54\sqrt{18} + 4 = 61.44 \text{ inches. Ans.}$$

EXAMPLE.—For what horsepower of boilers will a chimney 64 inches square and 125 feet high furnish draft?

SOLUTION.—By simply referring to Table 38, the H. P. is found to be 934.

1828. Chimneys are usually built of brick, though in some cases iron stacks are preferred. The external diameter of the base should be $\frac{1}{10}$ the height in order to provide stability. The taper of a chimney is from $\frac{1}{16}$ to $\frac{1}{4}$ inch to the foot on each side. The thickness of brickwork is usually one brick (8 or 9 inches) for 25 feet from the top, increasing $\frac{1}{2}$ brick for each additional 25 feet from the top downwards. If the inside diameter is greater than 5 feet, the top length should be $1\frac{1}{2}$ bricks, and if under 3 feet, it may be $\frac{1}{2}$ brick in thickness for the first 10 feet.

A round chimney is better than a square one, and a straight flue better than a tapering one. If the flue is tapering, the area for calculation is measured at the top.

The flue through which the gases pass from the furnaces to the chimney should have an area equal to or a little larger than the area of the chimney. Abrupt turns in the flue, or contractions of its area, should be carefully avoided, as they greatly retard the flow of the gases. Where one chimney serves several boilers, the branch flue from each furnace to the main flue must be somewhat larger than its proportionate part of the area of the main flue.

1829. The *maximum* rates of combustion attainable under natural draft are given by the following formulas,

which have been deduced from the experiments of Isherwood:

Let F = weight in pounds of coal per hour per square foot of grate area;

H = height in feet of chimney or stack.

Then, for anthracite coal burned under the most favorable conditions,

$$F = 2\sqrt{H} - 1; \qquad \textbf{(198.)}$$

and under ordinary conditions,

$$F = 1.5\sqrt{H} - 1. \qquad \textbf{(199.)}$$

For best semianthracite and bituminous coals,

$$F = 2.25\sqrt{H}; \qquad \textbf{(200.)}$$

and for less valuable soft coals,

$$F = 3\sqrt{H}. \qquad \textbf{(201.)}$$

The maximum rate of combustion is thus fixed by the height of the chimney; the minimum rate may be anything less. The customary rates have been given in Art. **1803.**

EXAMPLE.—Under ordinary conditions, what is the maximum rate of combustion of anthracite coal if the chimney is 120 feet high?

SOLUTION.—By formula **199,**

$F = 1.5\sqrt{120} - 1 = 15\frac{1}{2}$ lb. per sq. ft. per hr. Ans.

EXAMPLES FOR PRACTICE.

1. What should be the height of a chimney to give a draft pressure of $\frac{5}{8}$ inch of water, the temperature of the air being 60° F. and of the gases 440° F.? Ans. 107 ft.

2. A chimney is 135 feet high and 5 feet square inside. Calculate the horsepower for which it will furnish draft. Ans. 851 H. P.

3. What is the maximum rate of combustion of best bituminous coal in a marine boiler with chimney stacks 100 feet high? Ans. 22.5 lb.

4. Calculate the power required to furnish a forced draft at a pressure equal to $2\frac{1}{2}$ inches of water for a grate area of 100 sq. ft., the rate of combustion being 60 lb. per sq. ft. per hour, and the volume of air 200 cu. ft. per min. for each pound of coal. Assume the efficiency of the apparatus to be 70 per cent. Ans. $11\frac{1}{4}$ H. P.

5. Calculate the side of a square chimney 150 feet high, which furnishes draft for boilers of 1,000 H. P. Ans. 63.4 in.

6. What draft pressure will a chimney 80 feet high furnish, the temperatures of the air and gases being, respectively, 60° and 600° F.? Ans. .57 in.

7. Under the most favorable conditions, what height of chimney will allow a maximum rate of combustion of anthracite coal of 23 lb. per sq. ft. of grate per hour? Ans. 144 ft.

PROPORTIONS OF BOILERS.

1830. The **grate area** depends upon the rate of combustion, the evaporation per pound of fuel, and the total weight of steam evaporated per hour. For example, a 600 H. P. boiler plant generates per hour $600 \times 30 = 18,000$ pounds of steam having a pressure of 70 pounds from a feed-water temperature of 100°.

Supposing the rate of combustion to be 16 pounds of coal per square foot of grate per hour, and the evaporation to be 9 pounds of water per pound of coal, the necessary grate area must be $\frac{18,000}{16 \times 9} = 125$ square feet. This grate area would necessarily be divided between several furnaces, as a grate longer than 6 feet cannot be easily fired, while the width is limited to 4 or 6 feet by constructive considerations. Assuming the dimensions of the grate to be 4×6 feet, the above grate area would require $\frac{125}{4 \times 6} = 5$ furnaces.

From the above method of calculating grate area, the following formula may be readily deduced:

Let G = area of grate in square feet;
F = rate of combustion in pounds per square foot of grate surface per hour;
W = weight of steam generated per hour by boiler or boilers;
e = evaporation in pounds of water per pound of coal per hour.

Then, $$G = \frac{W}{Fe}. \qquad \textbf{(202.)}$$

In using this formula care must be taken that W and e are taken at the same pressure and temperature. That is, if the boiler generates W pounds of steam per hour at a pressure of 80 lb. from a temperature of 60°, e must represent the number of pounds of water which a pound of coal will raise from 60° and evaporate into steam at 80 lb. pressure. As will be explained later, it is customary, for the purpose of calculation, to reduce both W and e to the equivalent evaporation from and at 212° F.

EXAMPLE.—Find the grate area of an 80 H. P. boiler, evaporating 30 lb. of water from and at 212° per H. P. per hour, the rate of combustion being 12 lb. per sq. ft. of grate surface per hr., and the evaporation $10\frac{1}{2}$ lb. of water from and at 212° per pound of coal.

SOLUTION.—Using formula **202**, we obtain

$$G = \frac{W}{Fe} = \frac{80 \times 30}{12 \times 10.5} = 19 \text{ sq. ft.} \quad \text{Ans.}$$

1831. The **heating surface** of a boiler includes the entire surface of the shell and flues coming in contact with the flame and furnace gases on one side and water on the other; this includes, in the case of externally fired boilers, the portion of the shell below the fire line, portions of the heads, and the inner surface of fire tubes and flues, or the outer surface of water tubes. In the case of internally fired boilers, the heating surface includes the interior of the firebox, or furnace flues, and the inner surface of the tubes, if there are any.

The area of the heating surface of each of the various types of boilers bears a nearly constant ratio to the grate area. The ratios usually adopted are as follows:

Plain cylindrical boilers	12 to 15
Cornish	15 to 30
Cylindrical flue	20 to 25
Cylindrical tubular	25 to 35
Marine fire tubular	30 to 35
Marine water tubular	35 to 40
Locomotive tubular	50 to 100

1832. From a large number of tests of horizontal tubular boilers, Mr. G. H. Barrus concludes that the ratio of heating surface to the grate area for that type of boiler should be 36 to 1, where the rate of combustion is not above 12 lb. per sq. ft. of grate; this applies to anthracite coal. For bituminous coal the ratio may be 45 to 50 to 1. He also finds that the highest efficiency with anthracite coal is obtained when the combined cross-sectional area of the tubes is from $\frac{1}{9}$ to $\frac{1}{10}$ the grate area, and the greatest efficiency is obtained with bituminous coal when the tube area is from $\frac{1}{4}$ to $\frac{1}{7}$ the grate area.

1833. Since the greater part of the heating surface of tubular, and especially of locomotive, boilers is furnished by the tubes, particular attention must be paid to their arrangement. The tubes are from $2\frac{1}{2}$ to 4 inches in diameter; their length should not exceed 5 feet per inch of diameter, and a less length would probably be an advantage. The tubes should be arranged in horizontal and vertical rows; if staggered, they hinder the circulation of the water. The pitch of the tubes—that is, the distance from center to center—should be from $1\frac{1}{3}$ to $1\frac{1}{2}$ times the diameter of the tube; and the distance between the two center rows should be double the distance between the other rows. The distance between the shell and outer row of tubes should be 3 inches or more. It is good practice not to carry the tubes down near the bottom of the shell, as the lower tubes receive only the coolest gases, and do not, therefore, furnish efficient heating surface. By leaving them out a large body of water rests directly over the fire and a good circulation is insured. The distance from the top of the tubes to top of the shell should be about $\frac{2}{5}$ the diameter of the shell; if steam drums or domes are used this height may be $\frac{3}{8}$ or $\frac{1}{3}$ the diameter of the shell.

1834. The calculation of the heating surface of boiler tubes may be facilitated by reference to the following table:

TABLE 39.

LAP-WELDED AMERICAN CHARCOAL IRON BOILER TUBES.

STANDARD DIMENSIONS.

External Diameter.	Standard Thickness.	Wire Gauge.	Internal Diameter.	Internal Circumference.	External Circumference.	Length of Tube per sq. ft. Inside Surface.	Length of Tube per sq. ft. Out. Surface.	Internal Area.	External Area.	Weight per Foot.
Inches	Inches.		Inches.	Inches.	Inches.	Feet.	Feet.	Sq. in	Sq. in.	Lb
1	.072	15	.856	2.689	3.142	4.463	3.819	.575	.785	.708
1¼	.083	14	1.334	4.191	4.712	2.863	2.547	1.396	1.767	1.250
2	.098	13	1.804	5.667	6.283	2.118	1.909	2.556	3.142	1.981
2½	.109	12	2.283	7.172	7.854	1.673	1.528	4.094	4.909	2.755
3	.109	12	2.783	8.743	9.425	1.373	1.273	6.083	7.069	3.333
3½	.119	11	3.262	10.248	10.996	1.171	1.091	8.357	9.621	4.272
4	.130	10	3.741	11.753	12.566	1.021	.955	10.992	12.566	5.320
4½	.130	10	4.241	13.323	14.137	.901	.849	14.126	15.904	6.010
5	.140	9.5	4.720	14.828	15.708	.809	.764	17.497	19.635	7.276
6	.151	9	5.699	17.904	18.850	.670	.637	25.509	28.274	9.346

EXAMPLE.—Calculate the heating surface of a boiler 5 ft. in diameter, 15 ft. long, and containing 82 3-inch tubes. The boiler is so set that half the shell is exposed to the fire.

SOLUTION.—The heating surface of the shell is $\frac{1}{2} \times 5 \times 3.1416 \times 15 = 117.8$ sq. ft.

The heating surface of each head is $\frac{1}{2}$ its area less the cross-sectional area of the tubes; the area of the tubes may be taken from Table 39. Hence, heating surface of heads $= 2\left[\frac{1}{2} \times 5^2 \times .7854 - \frac{82 \times 7.069}{144}\right] = 11.6$ sq. ft.

The total length of tubes is $82 \times 15 = 1,230$ ft., and from Table 39 the length per sq. ft. inside surface is 1.373 ft. Hence, the heating surface of tubes is $\frac{1,230}{1.373} = 896$ sq. ft.

The total heating surface is, therefore,

$$117.8 + 11.6 + 896 = 1,025.4 \text{ sq. ft.}$$

Since the front head or tube plate is an inefficient heating surface, some authorities do not include it in calculating the effective heating surface. Leaving it out in the above example, the heating surface will be 1,019.6 sq. ft.

1835. The **steam space** required by a given boiler depends upon the purpose for which the steam is required. Where the steam is under high pressure, and where relatively small quantities are withdrawn at very frequent intervals, as in locomotives, the steam space need not be so large as where large quantities are withdrawn less frequently. Where the boiler supplies steam to an engine it is a general rule that the steam space should hold enough steam to supply the engine for 20 to 25 seconds. When steam is supplied for heating purposes it need not necessarily be dry, and, hence, the steam space may be smaller. As ordinarily designed, from $\frac{1}{4}$ to $\frac{1}{3}$ the cubic contents of the boiler is steam space, the remainder water space.

1836. The proportions of horizontal tubular and vertical boilers adopted by a leading manufacturing company are given in the following table:

TABLE 40.

PROPORTIONS OF VERTICAL BOILERS.

Horsepower....		34	28	24	23	20	17	15	12	11	10	8	6.5	5	4
Diam. of shell..	Inches.	48	44	42	40	38	36	34	32	30	28	26	24	22	20
Height of boiler	Ft. & in.	10–8	11–4½	9–5½	9–8½	8–9¼	8–8½	8–7¼	7–6¼	7–6¾	7–8½	7–4¼	7–10¾	7–4¼	8–3½
Diam. of tubes.	Inches.	2½	2½	2	2	2	2	2	2	2	2	2	2	2	2
Length of tubes	Ft. & in.	7–8	8–5¼	6–7⅜	6–11¼	6–1	6–1¼	6–1½	5–1	5–2¼	5–5	5–2	5–9½	5–4	6–4½
No. of tubes...		91	64	96	85	85	73	61	61	55	42	37	26	22	14
Heating surf'ce	Sq. ft.	340	282	247	233	201	177	152	124	117	100	81	68	53	42
Diam. of grate.	Inches.	42½	38	36	34¼	32½	30½	28¼	26½	24½	23	21	19	17	15
Total grate area	Sq. ft.	9.722	8.08	7.068	6.398	5.760	5.073	4.352	3.83	3.343	2.885	2.335	1.967	1.527	1.227
Total tube area	Sq. ft.	2.584	1.8170	1.6902	1.5045	1.5045	1.2921	1.0797	1.0707	0.9735	0.7434	0.6540	0.4602	0.3804	0.2475

PROPORTIONS OF HORIZONTAL RETURN TUBULAR BOILERS.

Horsepower..................		120	110	100	90	80	70	60	50	40	30	20
Diameter....................	Inches.	72	72	66	66	60	60	54	54	48	48	40
Length......................	Feet.	20	18	18	16	16	15	16	15	14	12	10
Diameter of tubes............	Inches.	3½	3½	3	3	3	3	3	3	3	3	3
Number of tubes..............		95	95	102	102	86	82	62	56	44	40	28
Heating surface tubes..........	Sq. feet.	1622.6	1460.3	1336.6	1188	1001.7	895.4	722.1	611.5	451.7	349.4	205.8
Heating surface shell...........	Sq. feet.	202.9	184.1	169.3	150	135.1	127.4	121.2	114.5	94.6	82.3	57.3
Heating surface, total.......	Sq. feet.	1825.5	1644.4	1505.9	1338	1136.8	1022.8	843.3	726	546.3	431.7	263.1
Grate area....................	Sq. feet.	52	47	43	38	32	29.2	25.6	20.7	15.6	12.3	7.5
Length of grate................	Ft., in.	8	8	7–2	6–4	6	5–6	5	4–7	3–11	3–2	2–2
Width of grate................	Ft., in.	6.5	6	6	6	5.25	5.25	5.125	4.5	4	4	3.5

HORSEPOWER OF BOILERS.

1837. The **horsepower** of a boiler is a measure of its capacity for generating steam. Boiler-makers usually rate the horsepower of their boilers as a certain fraction of the heating surface. Thus, in Table 40, just given, it will be noticed that for the vertical boilers the horsepower is $\frac{1}{10}$ the heating surface, and for the horizontal type, it is about $\frac{1}{15}$ the heating surface. The ratio of heating surface to horsepower, when the boiler is run under ordinary conditions, is about as follows:

Type of Boilers.	Sq. ft. of Heating Surface per Horsepower.
Water tube	10 to 12
Multitubular	14 to 18
Flue	8 to 12
Plain cylindrical	6 to 10
Locomotive (stationary practice)	12 to 16

The above method of rating boilers is extremely indefinite; with the same heating surface, different boilers of the same type may, under different circumstances, generate very different quantities of steam.

1838. In order to have an accurate standard of boiler power, the American Society of Mechanical Engineers has adopted as a commercial horsepower *an evaporation of 30 pounds of water per hour from a feed-water temperature of 100° F. into steam at 70 pounds gauge pressure*, which is considered equivalent to $34\frac{1}{2}$ units of evaporation; that is, to $34\frac{1}{2}$ pounds of water evaporated from a feed-water temperature of 212° F. into steam at the same temperature.

Since 966.1 heat units are required to evaporate a pound of water from and at 212°, a horsepower is equal to $966.1 \times 34\frac{1}{2} = 33,330$ B. T. U. per hour.

Different boilers generate steam at different pressures and receive the feed-water at different temperatures; in order to compare the relative performances of boilers it is necessary to reduce the actual evaporation to *an equivalent evaporation from and at 212° F. per pound of combustible:*

that is, what the evaporation would have been if the fuel had been without ash, the feed-water at 212° F., and the steam delivered at 0 gauge pressure.

Let W = the actual evaporation;
H = total heat of steam above 32° at pressure of actual evaporation;
t = observed temperature of feed-water;
W' = equivalent evaporation from and at 212°.

Then, $W(H - t + 32) = 966.1\, W'$,

$$\text{or } W' = \frac{W(H - t + 32)}{966.1}. \qquad \textbf{(203.)}$$

EXAMPLE.—A boiler generates 2,200 lb. of steam per hour at a pressure of 120 lb.; the temperature of the feed-water is 70° F. (*a*) What is the equivalent evaporation, and (*b*) the horsepower of the boiler?

SOLUTION.—(*a*) From the steam table, the total heat H, corresponding to a pressure of 120 lb., gauge, is 1,188.64 B. T. U. Hence, from formula **203**,

$$W' = \frac{2,200(1,188.64 - 70 + 32)}{966.1} = 2,620 \text{ lb. Ans.}$$

(*b*) The horsepower $= \frac{2,620}{34.5} = 76$ H. P., nearly. Ans.

1839. The quantity $\left(\frac{H - t + 32}{966.1}\right)$ which changes actual evaporation to equivalent evaporation from and at 212° is called the **factor of evaporation.**

To facilitate the calculation of the equivalent evaporation, the following table of factors of evaporation is inserted:

TABLE 41.

FACTORS OF EVAPORATION.

Temperature of Feed-water.	Gauge Pressures.															
	25	30	35	40	45	50	60	70	80	90	100	120	140	160	180	200
	Factors of Evaporation.															
32	1.204	1.206	1.209	1.211	1.212	1.214	1.217	1.219	1.222	1.224	1.227	1.231	1.234	1.237	1.239	1.241
40	1.196	1.198	1.201	1.203	1.204	1.206	1.209	1.211	1.214	1.216	1.219	1.223	1.226	1.229	1.231	1.233
50	1.185	1.187	1.190	1.192	1.193	1.195	1.198	1.200	1.203	1.205	1.208	1.212	1.215	1.218	1.220	1.222
60	1.175	1.177	1.180	1.182	1.183	1.185	1.188	1.190	1.193	1.195	1.198	1.202	1.205	1.208	1.210	1.212
70	1.165	1.167	1.170	1.172	1.173	1.175	1.178	1.180	1.183	1.185	1.188	1.192	1.195	1.198	1.200	1.202
80	1.154	1.156	1.159	1.161	1.162	1.164	1.167	1.169	1.172	1.174	1.177	1.181	1.184	1.187	1.189	1.191
90	1.144	1.146	1.149	1.151	1.152	1.154	1.157	1.159	1.162	1.164	1.167	1.171	1.174	1.177	1.179	1.181
100	1.134	1.136	1.139	1.141	1.142	1.144	1.147	1.149	1.152	1.154	1.157	1.161	1.164	1.167	1.169	1.171
110	1.123	1.125	1.128	1.130	1.131	1.133	1.136	1.138	1.141	1.143	1.146	1.150	1.153	1.156	1.158	1.160
120	1.113	1.115	1.118	1.120	1.121	1.123	1.126	1.128	1.131	1.133	1.136	1.140	1.143	1.146	1.148	1.150
130	1.102	1.104	1.107	1.109	1.110	1.112	1.115	1.117	1.120	1.122	1.125	1.129	1.132	1.135	1.137	1.139
140	1.092	1.094	1.097	1.099	1.100	1.102	1.105	1.107	1.110	1.112	1.115	1.119	1.122	1.125	1.127	1.129
150	1.082	1.084	1.087	1.089	1.090	1.092	1.095	1.097	1.100	1.102	1.105	1.109	1.112	1.115	1.117	1.119
160	1.071	1.073	1.076	1.078	1.079	1.081	1.084	1.086	1.089	1.091	1.094	1.098	1.101	1.104	1.106	1.108
170	1.061	1.063	1.066	1.068	1.069	1.071	1.074	1.076	1.079	1.081	1.084	1.088	1.091	1.094	1.096	1.098
180	1.050	1.052	1.055	1.057	1.058	1.060	1.063	1.065	1.068	1.070	1.073	1.077	1.080	1.083	1.085	1.087
190	1.040	1.042	1.045	1.047	1.048	1.050	1.053	1.055	1.058	1.060	1.063	1.067	1.070	1.073	1.075	1.077
200	1.030	1.032	1.035	1.037	1.038	1.040	1.043	1.045	1.048	1.050	1.053	1.057	1.060	1.063	1.065	1.067
210	1.020	1.022	1.025	1.027	1.028	1.030	1.033	1.035	1.038	1.040	1.043	1.047	1.050	1.053	1.055	1.057

EXAMPLE.—A boiler is required to furnish 1,800 lb. of steam per hour at a gauge pressure of 80 lb. If the temperature of the feed-water be 48°, what will be the rated horsepower of the boiler?

SOLUTION.—From Table 41, the factor of evaporation for 80 lb. pressure and a feed-water temperature of 40° is 1.214, and for the same pressure and a feed-water temperature of 50°, 1.203. Difference $= 1.214 - 1.203 = .011$. Difference of temperature $= 50° - 40° = 10°$. Difference between the lower temperature and the required temperature $= 48° - 40° = 8°$. Then, $10° : 8° :: .011 : x$, or $x = .009$. $1.214 - .009 = 1.205$. $1,800 \times 1.205 = 2,169$ lb., and $\frac{2169}{34.5} = 63$ horsepower, nearly. Ans.

EXAMPLE.—What is the factor of evaporation when the feed-water temperature is 122° and the gauge pressure 72 lb.?

SOLUTION.—In Table 41, under the column headed 70 and opposite 120 in left-hand column is found 1.128; in column headed 80 and opposite 120 is found 1.131; difference $= .003$. In the same vertical columns and opposite 130 are found 1.117 and 1.120; difference $= .003$, same as above. Hence, for an increase of 10 lb. in gauge reading, there is an increase of .003 in the factor of evaporation, or an increase of .0003 for 1 lb. and of $.0003 \times 2 = .0006$ for 2 lb. Therefore, for a feed-water temperature of 120° and 72 lb. pressure, the factor of evaporation is $1.128 + .0006 = 1.1286$.

The difference between the numbers opposite 120 and 130 in the two columns headed 70 and 80, respectively, is $1.128 - 1.117 = .011$, and $1.131 - 1.120 = .011$, showing that, for an increase of temperature in the feed-water of 10°, there is a decrease in the factor of .011, and for 1° a decrease of .0011, or for 2° of .0022. Hence, the value of the factor for a temperature of 122° and a pressure (gauge) of 72 lb. is $1.1286 - .0022 = 1.1264$, using four decimal places. Ans.

BOILER TRIALS.

1840. The **object** of a boiler trial may be to determine: the efficiency of the boiler under given conditions; the comparative value of different boilers working under the same conditions; the comparative value of fuels; the evaporative power or the horsepower of the boiler; or the quantity of coal and steam consumed by a given engine.

1841. The **efficiency** of the boiler may be defined as the ratio of the heat utilized in evaporating water to the total heat supplied by the fuel. The amount of heat

supplied is determined by accurately weighing the fuel used during the test, and deducting all the ash and unconsumed portions. The total heat of combustion of the fuel obtained by experiment or calculation, multiplied by the quantity of combustible, should give the heat units supplied by the combustion, assuming it to be perfect. The heat utilized in useful evaporation is obtained by weighing the feed-water delivered to the boiler and multiplying this weight by the number of heat units required to change water at the temperature of the feed into steam at the observed pressure.

The essential operations of a boiler trial are the weighing of the feed-water and fuel, and the observations of the steam pressure, temperature of feed-water, and various other less important pressures and temperatures.

1842. Method of Making a Boiler Trial.—A standard method of making boiler trials has been adopted by the American Society of Mechanical Engineers. The method is as follows :

CODE OF RULES FOR BOILER TRIALS.

1843. I. **Preliminaries to a Test.**—In preparing for and conducting trials of steam boilers, the specific object of the proposed trial should be clearly defined and steadily kept in view.

II. Measure and record the dimensions, position, etc., of grate and heating surfaces, flues, and chimneys; proportion of air space in the grate surface; kind of draft, natural or forced.

III. Put the boilers in good condition. Have heating surfaces clean inside and out, grate bars and sides of furnace free from clinkers, have dust and ashes removed from back connections, leaks in masonry stopped, and all obstruction to draft removed. See that the damper will open to full extent, and that it may be closed when desired. Test for leaks in masonry by firing a little smoky fuel and immediately closing damper. The smoke will then escape through the leaks.

IV. Have an understanding with the parties in whose interest the test is to be made as to the character of the coal to be used. The coal must be dry, or, if wet, a sample must be dried carefully and a determination of the amount of moisture in the coal made, and a calculation of the results of the test corrected accordingly. Wherever possible, the test should be made with standard coal of a known quality. For that portion of the country east of the Alleghany Mountains, good anthracite egg coal or Cumberland semi-bituminous coal may be taken as the standard for making tests. West of the Alleghany Mountains and east of the Missouri River, Pittsburg lump coal may be used.

V. In all important tests a sample of coal should be selected for chemical analysis.

VI. Establish the correctness of all apparatus used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.

2. Tanks or water meters for measuring water. Water meters should be used as a rule as a check on other measurements. For accurate work the water should be weighed or measured in a tank.

3. Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc.

4. Pressure gauges, draft gauges, etc.

VII. Before beginning a test, the boiler and chimney should be thoroughly heated to their usual working temperature. If the boiler is new it should be in continuous use at least a week before testing, so as to dry the mortar thoroughly and heat the walls.

VIII. Before beginning a test, the boiler and connections should be free from leaks, and all water connections, including blow-off and extra feed-pipes, should be disconnected or stopped with blank flanges, except the particular pipe through which the water is to be fed to the boiler during the trial. In locations where the reliability of power is so important that an extra feed-pipe must be kept in position, and, in general, when for other reasons water pipes other

than feed-pipes cannot be disconnected, such pipes may be drilled so as to leave openings in their lower sides, which should be kept open throughout the test as a means of detecting leaks or accidental or unauthorized opening of valves. During the test the blow-off pipe should remain exposed. If an injector is used it must receive steam directly from the boiler being tested, and not from a steam pipe or any other boiler.

See that the steam pipe is so arranged that water of condensation can not run back into the boiler. If the steam pipe has such an inclination that the water of condensation from any portion of the steam-pipe system may run back into the boiler, it must be trapped so as to prevent this water getting into the boiler without being measured.

1844. IX. **Starting and Stopping a Test.**—A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable. The conditions of the boiler and furnace should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same, the water-level the same, the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. To secure as near an approximation as possible to exact uniformity in conditions of the fire and in the temperatures of the walls and flues, the following method of starting and stopping a test should be adopted:

1845. X. **Standard Method.**—Steam being raised to the working pressure, remove rapidly all the fire from the grate, and as quickly as possible start a new fire with weighed wood and coal, noting the time of starting the test and the height of the water-level while the water is in a quiescent state just before starting the fire.

At the end of the test, remove the whole fire, clean the grates and ash pit, and note the water-level when the water is in a quiescent state; record the time of hauling the fire at the end of the test. The water-level should be as nearly

as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation and not by operating the pump after the test is completed. It will generally be necessary to regulate the discharge of steam from the boiler, tested by means of the stop-valve, for a time when fires are being hauled at the beginning or at the end of the test, in order to keep the steam pressure in the boiler at those times up to the average during the test.

1846. XI. **Alternate Method.**—Instead of the standard method above described, the following may be employed where local conditions render it necessary:

At the regular time for slicing and cleaning fires, have them burned rather low, as is usual before cleaning, and then thoroughly cleaned; note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the height of the water-level—which should be at the medium height to be carried throughout the test—at the same time; and note this time as the time of starting the test. Fresh coal which has been weighed should now be fired. The ash pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire and in the same condition on the grates as at the start. The water-level and steam pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

1847. XII. **During the Test.—Keep the Conditions Uniform.**—The boiler should be run continuously without stopping for meal-times or for rise or fall of steam due to change of demand for steam. The draft being adjusted to the rate of evaporation or combustion desired before the test is begun, it should be retained constant during the test by means of the damper.

If the boiler is not connected to the same steam pipe with other boilers, an extra outlet for steam, with valve in same,

should be provided, so that in case the pressure should rise to that at which the safety valve is set, it may be reduced to the desired point by opening the extra outlet, without checking the fires.

If the boiler is connected to the main steam pipe with other boilers, the safety valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in pressure the other boilers may blow off and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual.

All the conditions should be kept as nearly uniform as possible, such as the force of draft, pressure of steam, and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion and the kind of grates. When very good coal is used and the combustion is not too rapid, a ten-hour test may be run without any cleaning of the grates other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

1848. XIII. **Keeping the Records.** — The coal should be weighed and delivered to the firemen in equal portions, each sufficient for about one hour's run, and a fresh portion should not be delivered until the previous one has been all fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing each new portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired, at the end of the test, to discover the degree of uniformity of combustion, evaporation, and economy at different stages of the test.

1849. XIV. **Priming Tests.**—In all tests in which accuracy of results is important, calorimetric tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent., and the final records of the boiler test should be corrected according to the average results of the calorimeter tests.

On account of the difficulty of securing accuracy in these tests, the greatest care should be taken in the measurement of weights and temperatures. The thermometer should be accurate to within a tenth of a degree, and the scales on which the water is weighed to $\frac{1}{100}$ of a pound.

1850. XV. **Analysis of Gases.**—The analysis of gases and measurement of the air supply are not generally necessary in ordinary commercial tests; they are, however, valuable in determining the relative values of different methods of firing or of different kinds of furnaces. Care must be taken to procure average samples of the gas, and the analysis should be made by a competent chemist.

1851. XVI. **Record of the Test.**—A "log" of the test should be kept on properly prepared blanks containing headings as follows:

Time.	Pressures.			Temperatures.					Fuel.		Feed-Water.	
	Barometer.	Steam Gauge.	Draft Gauge.	External Air.	Boiler Room.	Flue.	Feed-Water.	Steam.	Time.	Pounds.	Time.	Pounds or Cubic Feet.

1852. XVII. **Reporting the Trial.**—The final results should be recorded upon a properly prepared blank, and should include as many of the following items as are adapted for the specific object for which the trial is made. The items marked with a * may be omitted for ordinary trials, but are desirable for a comparison with similar data from other sources.

Results of the trial of
Boilers at
To determine

	ITEMS.	
1.	Date of trial.	
2.	Duration of trial.	hours.
	DIMENSIONS AND PROPORTIONS.	
3.	Grate surface, wide long area	sq. ft.
4.	Water-heating surface.	sq. ft.
5.	Superheating surface.	sq. ft.
6.	Ratio of water-heating surface to grate surface.	
	AVERAGE PRESSURES.	
7.	Steam pressure in boiler by gauge.	lb.
*8.	Absolute steam pressure.	lb.
*9.	Atmospheric pressure per barometer.	in.
10.	Force of draft in inches of water.	in.
	AVERAGE TEMPERATURES, FAHR.	
*11.	Temperature of external air.	deg.
*12.	Temperature of fireroom.	deg.
*13.	Temperature of steam.	deg.
14.	Temperature of escaping gases.	deg.
15.	Temperature of feed-water.	deg.
	FUEL.	
15½.	Cost of coal per 2,000 lb. at boilers.	
16.	Total amount coal consumed (includes wood × 0.4).	lb.
17.	Moisture in coal.	per cent.
18.	Dry coal consumed.	lb.

19.	Total refuse dry .... pounds =	per cent.
20.	Total combustible (Item 18 less Item 19).	lb.
*21.	Dry coal consumed per hour.	lb.
*22.	Combustible consumed per hour.	lb.
	RESULTS OF CALORIMETRIC TESTS.	
23.	Quality of steam (dry steam taken as unity).	per cent.
24.	Percentage of moisture in steam.	per cent.
25.	Number of degrees superheated.	deg.
	WATER.	
26.	Total weight of water pumped into boiler and apparently evaporated.	lb.
27.	Water actually evaporated corrected for quality of steam.	lb.
28.	Equivalent water evaporated into dry steam from and at 212° F.	lb.
*29.	Equivalent total heat derived from fuel.	B. T. U.
30.	Equivalent water evaporated into dry steam from and at 212° F. per hour.	lb.
	ECONOMIC EVAPORATION.	
31.	Water actually evaporated per pound of dry coal from actual pressure and temperature.	lb.
31½.	Equivalent water evaporated for $1.00 from and at 212° F.	lb.
32.	Equivalent water evaporated per pound of dry coal from and at 212° F.	lb.
33.	Equivalent water evaporated per pound of combustible from and at 212° F.	lb.
	COMMERCIAL EVAPORATION.	
34.	Equivalent water evaporated per pound of dry coal with one-sixth refuse at 70 pounds gauge pressure from temperature of 100° F. (= Item 33 × .07249).	lb.
	RATE OF COMBUSTION.	
35.	Dry coal actually burned per square foot of grate surface per hour.	lb

*36.	Consumption of dry coal per hour, coal assumed with one-sixth refuse.	Per square foot of grate surface.	lb.
*37.		Per sq. ft. of water-heating surface.	lb.
*38.		Per sq. ft. of least area for draft.	lb.
	RATE OF EVAPORATION.		
39.	Water evaporated from and at 212° F. per sq. ft. of heating surface per hr.		lb.
*40.	Water evaporated per hour from temperature of 100° F. into steam of 70 lb. gauge pressure.	Per sq. ft. of grate surface.	lb.
*41.		Per sq. ft. of water-heating surface.	lb.
*42.		Per sq. ft. of least area for draft.	lb.
	COMMERCIAL HORSEPOWER.		
43.	On a basis of 30 pounds water per hour evaporated from a temperature of 100° F. into steam of 70 pounds gauge pressure (= $34\frac{1}{2}$ pounds from and at 212° F.).		H. P.
$43\frac{1}{2}$.	Number of horsepower obtained for $1.		H. P.
44.	Horsepower, builders' rating at . . . square feet per H. P.		H. P.
45.	Per cent. developed above or below rating.		per cent.

Some of the items of the above schedule are derived from the others as follows:

Item 27 = Item 26 × Item 23.

Item 28 = Item 27 × Factor of evaporation (taken from Table 40).

Item 29 = Item 28 × 966.1.

Item 31 = Item 27 ÷ Item 18.

Item 32 = Item 28 ÷ Item 18 = Item 31 × Factor of evaporation.

Item 33 = Item 28 ÷ Item 20.

Item 43 = Item 30 ÷ $34\frac{1}{2}$.

1853. The **quality of the steam** (Item 23) must be determined by the use of a calorimeter. The barrel calorimeter is quite commonly used for this purpose, though great care must be exercised in operating it, if trustworthy results are to be obtained. The method of procedure is as follows: A barrel or tank T, Fig. 577, holding 400 or 500 pounds of water, is placed on a platform scale S, filled with water and weighed. The temperature of the water is registered by a thermometer inserted in the side of the barrel. Steam from the boiler is led through a pipe or hose U, into the barrel until the temperature of the water reaches 130° to 140° F. The steam is then turned off, and the barrel and contents are again weighed. The average steam pressure throughout the observation must be observed. It is also well to have the tube bent as shown in the figure.

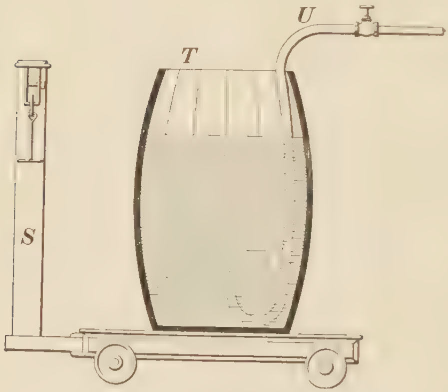


FIG. 577.

Knowing now the weight of the cold water in the barrel, the weight of steam run in, the initial and final temperatures of the water, and the steam pressure, the quality of the steam may be readily found as follows:

Let W = original weight of water in barrel;
w = weight of steam and water condensed in the cold water;
l = latent heat of steam at observed pressure;
t = temperature of steam at observed pressure;
t_1 = initial temperature of water;
t_2 = final temperature of water;
x = the portion of w which is dry steam,
Q = quality of steam; that is, the percentage of the mixture evaporated, which is pure dry steam.

The x pounds of dry steam give up $x\,l$ heat units on changing from steam to water; the combined steam and water w becomes lowered in temperature from $t°$ to $t_2°$ upon becoming mixed with the cold water in the barrel. Hence, the total number of heat units given up to the water in the barrel is

$$x\,l + w\,(t - t_2),$$

and this heat raises the temperature of the cold water from $t_1°$ to $t_2°$. Hence,

$$x\,l + w(t - t_2) = W(t_2 - t_1),$$

or $$x = \frac{W(t_2 - t_1) - w(t - t_2)}{l},$$

and $$Q = \frac{x}{w} = \frac{1}{l}\left[\frac{W}{w}(t_2 - t_1) - (t - t_2)\right]. \qquad \textbf{(204.)}$$

If Q is greater than 100 per cent. the steam is superheated, the amount of superheat being

$$\frac{(Q - 1)\,l}{0.48} \text{ degrees F.}$$

EXAMPLE.—In a calorimetric test, the weight of cold water was 420 pounds, of steam condensed 36 pounds. The initial temperature of cold water was 40° F., the final temperature was 130° F., and the steam pressure was 60 pounds. Find the quality of the steam.

SOLUTION.—By formula **204,**

$$Q = \frac{1}{l}\left[\frac{W}{w}(t_2 - t_1) - (t - t_2)\right] =$$

$$\frac{1}{898.5}\left[\frac{420}{36}(130 - 40) - (307.2 - 130)\right] = 97.14 \text{ per cent.};$$

that is, the boiler generates a mixture which is composed of 97.14 per cent. dry steam and 2.86 per cent. water. Ans.

The one point in favor of the barrel calorimeter is its availability. A barrel, platform scale, a length of hose, and a fairly good thermometer can be procured without trouble or great expense.

1854. For refined measurements of the quality of steam, more accurate instruments must be used. The greatest care must be exercised in reading the thermometer (which, for this purpose, should be so divided as to read tenths of a degree) and in weighing the water. The scales

should be as finely graduated as possible, also the gauge which records the steam pressure. If possible, a barometer should be used to obtain the exact atmospheric pressure which should be added to the gauge reading in order to obtain the absolute pressure for use in determining l and t when the steam tables are employed. Slight errors in weighing and reading the thermometer and gauges will make a considerable difference in the value of Q. For example, suppose that the observed readings were $W = 200.5$ lb., $w = 9.9$ lb., steam pressure (gauge) $p = 78$ lb., $t_1 = 44.5°$, and $t_2 = 100.5°$. Suppose, further, that the true readings should have been $W = 200$ lb., $w = 10$ lb., $p = 80$ lb., $t_1 = 45°$, and $t_2 = 100°$. Substituting in formula **204**,

		% moisture	% error
the true readings, the value of	$Q = .9874 =$	1.26	$= 0.00$
For all readings true except $W = 200.5$,	$Q = .9906 =$	0.94	$= 0.32$
For all readings true except $w = 9.9$,	$Q = 1.0000 =$	0.00	$= 1.26$
For all readings true except $p = 7.8$,	$Q = .9880 =$	1.20	$= 0.06$
For all readings true except $t_1 = 44.5$,	$Q = .9989 =$	0.11	$= 1.15$
For all readings true except $t_2 = 100.5$,	$Q = .9994 =$	0.06	$= 1.20$
For all readings incorrect,	$Q = 1.0272 =$	-2.72	$= 3.98$

The last case indicates that the steam has been superheated 50.1°, since

$$\frac{(Q-1)\,l}{.48} = \frac{(1.0272)-1) \times 887.687}{.48} = 50.1°.$$

1855. The Separator Calorimeter.—Fig. 578, as designed by Prof. R. C. Carpenter, is perhaps as simple and reliable as any in use. It consists of a chamber A, into which is led the steam pipe S from the boiler. The lower end of this pipe is perforated with numerous $\frac{1}{8}$-inch holes. The exhaust pipe S' is connected with the top of the chamber. Across this exhaust pipe extends a diaphragm with a center orifice $\frac{1}{16}$ inch in diameter. As the steam enters through the pipe $B\ S$ its velocity is checked and the particles of entrained water are carried on to the bottom of the chamber A by their inertia, while the steam, now practically dry, passes out through the exhaust pipe S'. In conducting the test, the water-level in the beginning is

noted on the gauge glass C by tying a thread around it, or by some other simple means. The water separated from the steam is drawn off through the pipe W and weighed. The water-level at the end of the test is brought down to that at the beginning and hence the water withdrawn represents exactly the amount separated from the steam. A steam gauge may be attached to the pipe D if desired, although it is not necessary to know either the pressure or the temperature of the steam when using this instrument. The steam from the pipe S' is led into a tank of cold water and weighed. Hence, if

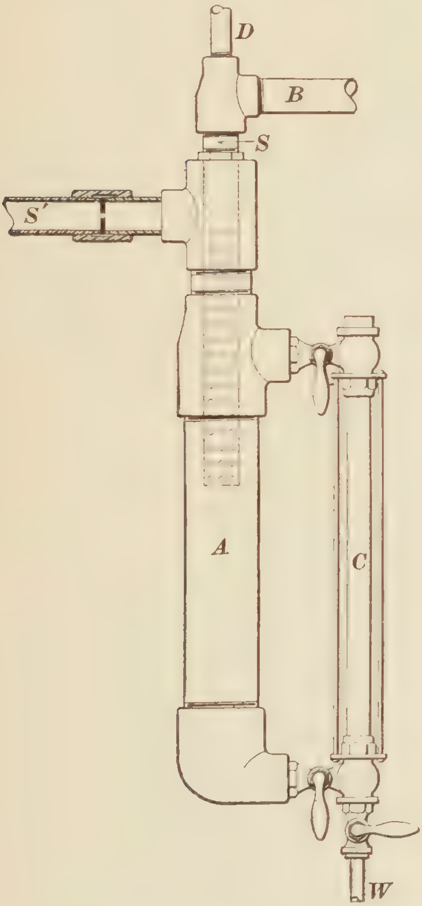


FIG. 578.

W = weight of dry steam discharged from pipe S';

w = weight of water drawn from separator;

R = water condensed by radiation;

and Q = quality of steam;

we have

$$Q = \frac{W+R}{W+w}. \quad \textbf{(205.)}$$

The radiation loss R may be determined by coupling two calorimeters together; since the first gives off dry steam, any water collecting in the second must be due to radiation.

Therefore, the water collected in the second calorimeter is the quantity R of formula **205.**

EXAMPLE.—In a calorimetric test, the dry steam condensed during the test was 8.5 pounds, the weight of water separated $\frac{1}{4}$ pound, and the radiation loss $\frac{1}{10}$ pound. What was the quality of the steam?

SOLUTION.—Formula **205** gives

$$Q = \frac{W+R}{W+w} = \frac{8.5+0.1}{8.5+.25} = 98\tfrac{2}{7} \text{ per cent. Ans.}$$

1856. In conducting a boiler trial, the various observations of temperatures, pressures, etc., should be made simultaneously at intervals of about fifteen minutes. The coal supplied to the furnace is weighed out in lots of 500 or 600 pounds. It is a convenient plan to have a box with one side open placed on a platform scale. A weight is then placed on the scale beam sufficient to balance the box. The scale may then be set at 500 or 600 pounds, the coal shoveled in until the beam rises, and then fed directly from the box to the furnace.

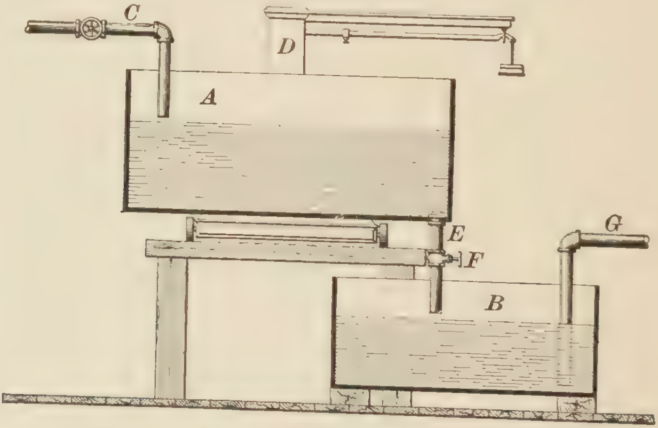


FIG. 579.

After the test, the ashes and clinkers must be raked from the ash pit and grate and weighed. This weight subtracted from the weight of the coal used gives the amount of combustible—Item 20.

1857. A convenient way to measure the feed-water delivered to the boiler is to have two tanks A and B, Fig. 579,

one above the other. The supply of water is fed into the upper tank through the pipe *C*, which rests on a platform scale *D*. After balancing the tank, the scale may be set to weigh 500 or 600 pounds of water, the water being run in until the beam rises, and then shut off. The upper tank is provided with a pipe *E* and valve *F*, by means of which the water may be discharged into the lower tank, from which it is fed to the boiler through the pipe *G*.

1858. The attendant who keeps the record of coal supply or water supply should become accustomed to making the tally on his blank just before or after some regular operation. For example, the person who weighs the feed-water should record each tankful, say immediately after closing the valve in the supply pipe, or, perhaps, after emptying the upper tank into the lower. If this precaution is not observed, the attendant is liable to become uncertain as to whether he has recorded the previous data, and a mistake is almost certain to result.

EXAMPLE.—Given the following data observed during a boiler trial; required to make the necessary calculations for economic evaporation and horsepower:

Duration of test	10 hr.
Average gauge pressure	72 lb.
Average temperature of feed-water	122° F.
Pounds of coal burned	15,232 lb.
Percentage of ash	4½%.
Water evaporated at temp. of feed	124,600 lb.
Average quality of steam	97.2%.
Rated horsepower	300.

CALCULATIONS.—Water evaporated $= 124{,}600 \times .972 = 121{,}111.2$ lb.

Water evaporated per pound of coal—actual conditions $= 121{,}111.2 \div 15{,}232 = 7.95$ lb.

Water evaporated per pound of combustible $= 121{,}111.2 \div (15{,}232 \times .95\frac{1}{2}) = 121{,}111.2 \div 14{,}546.6 = 8.33$ lb.

From Table 40, the factor of evaporation for the given pressure and temperature of feed-water is 1.1264.

Hence, the equivalent evaporation from and at 212° per pound of coal is $7.95 \times 1.1264 = 8.96$ lb.

The equivalent evaporation per pound of combustible is $8.33 \times 1.1264 = 9.38$ lb.

The total equivalent evaporation from and at 212° F. per hour is

$$\frac{121,111.2 \times 1.1264}{10} = 13,641.966 \text{ lb.}$$

The horsepower is, therefore,

$$13,641.966 \div 34\tfrac{1}{2} = 395.42 \text{ H. P.}$$

The per cent. above rated capacity is

$$\frac{395.42 - 300}{300} = 31.81 \text{ per cent.}$$

INCRUSTATION.

1859. Incrustation is the deposit formed in a boiler by the precipitation of mineral substances held in solution by the feed-water, or by the settling of mud or earthy matters which are held in suspension by the feed-water. The gradual accumulation of these foreign substances sometimes leads to the formation of a hard coating of scurf or scale, $\frac{1}{4}$ to $\frac{1}{2}$ inch thick, over the plates and tubes covered by water. A very thin scale is regarded by some engineers as beneficial, since it protects the plates from the corrosive action of the water; but when it becomes at all thick, the conducting power of the plates is largely decreased, it being stated that a scale $\frac{1}{16}$ inch thick demands an increase of 15 per cent. in fuel, and scales of greater thickness in proportion. A thick incrustation may become a source of danger, since the heat from the furnace is not carried away by the circulation of the water as fast as it would otherwise be, and hence the furnace plates are liable to become overheated or even burned through.

The most troublesome scale-forming substances are the carbonates and sulphates of lime and magnesia. Carbonate of lime (which is the same thing as marble or limestone) will dissolve in water which contains carbonic acid. By heating the water the carbonic acid is driven off, and the carbonate is precipitated in the solid form. The carbonate of magnesia acts in a similar manner. The sulphates are soluble in water at low temperatures, but are completely insoluble in water at a temperature of about 300° F. Thus,

it is evident that if the feed-water is heated to about 300°, the greater part of the scale-forming materials will be precipitated in the solid form. The incrustation formed by carbonate of lime is soft and granular, while that from the sulphate is hard and crust-like.

The presence of grease or organic matter, in combination with carbonate of lime, greatly aggravates the danger of overheating the plates, and makes the scale harder and more troublesome. For this reason only mineral oils should be used in engine cylinders, if the condensed steam is to be fed back to the boiler.

1860. Prevention of Incrustation.—Incrustation may be prevented: 1. By separating out the scale-forming substances before the water is fed into the boiler. 2. By removing the soft scale before it has had time to solidify and bake into a solid crust.

1861. The water may be freed from its impurities to a greater or less extent by passing it through a **purifier** before allowing it to enter the boiler. In the purifier the temperature is raised until the water will no longer hold the carbonates and sulphates in solution; they are therefore precipitated and remain in the purifier, while the water, now freed from them, passes on to the boiler. Where the water holds an excess of foreign matter in suspension, it may be advisable to filter it through beds of pebbles, bones, or other material.

1862. The readiest method of removing impurities after they are deposited in the boiler is by the blow-out apparatus. A large part of the scale is naturally carried to the coolest part of the boiler (to the mud drum, if there is one), and may be removed by blowing off the boiler while under steam pressure.

1863. The fact that many impurities are held in suspension and float as a scum on the water for some time

before settling has led to the use of the surface blow-out apparatus. The *Hotchkiss mechanical cleaner* is a form of surface blow-out apparatus.

It consists of a cast-iron spherical vessel situated on top of the boiler. This vessel is connected to each end of the boiler by a pipe. On one end of the pipe leading to the front end of the boiler (called the uptake pipe) is a large funnel, so placed as to be partially submerged in water. When the boiler is in operation, the natural circulation of the water causes it to rise in the uptake pipe and flow into the spherical vessel, the funnel scooping in the impurities floating on the top. The water then flows out through the downtake pipe into the rear, or cooler, end of the boiler. The water in the vessel being comparatively quiet, the impurities are deposited at the bottom and may be blown out at intervals.

A frequent use of both surface and bottom blow-outs will keep a boiler comparatively free from incrustation.

1864. Incrustation is prevented to a large extent by a rapid water circulation. This is one of the chief merits claimed for the water-tube boilers; the sediment is swept through the tubes and shell and deposited in the lowest part of the boiler—the mud drum.

1865. Various chemical substances are introduced into the boiler to combine with the scale-forming material and change its character. The cheapest and most effective of these substances is carbonate of soda. When the scale consists of sulphate of lime, the combination of the soda and sulphate results in the formation of sulphate of soda, which is soluble, and carbonate of lime, which forms a soft scale, which is easily blown off. Where the water contains carbonate of lime, sal ammoniac or caustic lime may be used to prevent a hard incrustation. Sometimes organic substances containing tannic acid, such as oak bark, hemlock, or sumac, are employed to loosen or prevent scale. They are liable to injure the plates by corrosion, and hence should not be

used. The following is a list of troublesome scale-forming substances and their remedies:

Troublesome Substances.	Trouble.	Remedy or Palliation.
Sediment, mud, clay, etc.	Incrustation.	Filtration. Blowing-off.
Readily soluble salts.	Incrustation.	Blowing-off.
Bicarbonates of lime, magnesia, iron.	Incrustation.	Heating feed. Addition of caustic soda, lime, or magnesia.
Sulphate of lime.	Incrustation.	Addition of carbonate of soda or barium chloride.
Chloride and sulphate of magnesium.	Corrosion.	Addition of carbonate of soda, etc.
Carbonate of soda in large amounts.	Priming.	Addition of barium chloride.
Acid (in mine water).	Corrosion.	Alkali.
Dissolved carbonic acid and oxygen.	Corrosion.	Heating feed. Addition of caustic soda, slacked lime, etc.
Grease (from condensed water).	Corrosion.	Slacked lime and filtering. Carbonate of soda. Substitute mineral oil.
Organic matter (sewage).	Priming.	Precipitate with alum or chloride of iron and filter.
Organic matter.	Corrosion.	Same as last.

1866. Zinc is largely used in marine boilers for the prevention of both incrustation and corrosion. The scale

may acquire thickness and hardness, but can easily be removed from the plates. It is supposed that the zinc in connection with the iron of the plates keeps up a feeble galvanic action, and that the hydrogen liberated at the surface of the plate by this action prevents the incrustation from adhering to it. The zinc is distributed through the boiler in the form of slabs. About one square inch of zinc surface should be supplied for every 50 pounds of water.

1867. Kerosene oil has been found useful in preventing and removing scale. It is claimed by those who have used it that one quart per day per 100 horsepower is sufficient to keep boilers free from scale, though using very hard and impure water. It is also effective in breaking up and loosening hard scale after it is formed.

The most certain and effective remedy for incrustation after it has been once deposited is to remove it mechanically at certain intervals. The boiler should be entered and the scale chipped off or pulled off by hand.

WEAR AND TEAR.

1868. Corrosion may be defined as the eating away or wasting of the plates due to the chemical action of impure water. It is probably the most destructive of the various forces which tend to shorten the life of the boiler. Corrosion is of two forms—internal and external. Internal corrosion may present itself as: 1, uniform corrosion; 2, pitting or honeycombing; 3, grooving.

1869. In cases of uniform corrosion, large areas of plate are attacked and eaten away. There is no sharp line of division between the corroded part and the sound plate, and oftentimes the only way of detecting the corrosion is to drill a hole through the suspected plate and thus ascertain its thickness. Corrosion often violently attacks the staybolts and rivet heads.

1870. Pitting and honeycombing are readily perceived. The plates are in spots indented with holes and cavities from

$\frac{1}{32}$ to $\frac{1}{4}$ inch deep. The appearance of a pitted plate is shown in Fig. 580.

1871. Grooving is generally caused by the buckling action of the plates when under pressure. Thus, the ordinary lap joint of a boiler distorts the shell slightly from a truly cylindrical form, and the steam pressure tends to bend

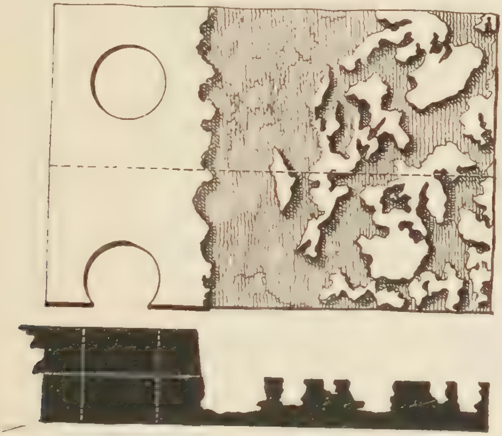


FIG. 580.

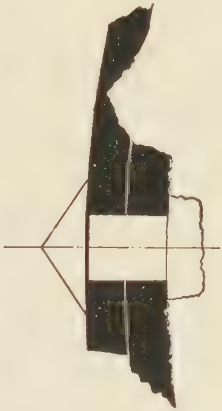


FIG. 581.

the plates at the joint. This bending action is liable to start a small crack along the lap, which being acted upon by corrosive agents in the water, soon deepens into a groove, as shown in Fig. 581. The mark made along the seam by the sharp calking tool, when used by careless workmen, is almost certain to lead to grooving.

1872. To prevent corrosion, the feed-water should be as free as possible from corrosive impurities. When bad water must be used, the corrosive impurities must be neutralized by adding alkaline substances, such as caustic soda or soda ash. The remedies for corrosive impurities were given along with those for incrustation.

Zinc is used to prevent corrosion in marine boilers. It is believed that corrosion is due in some measure to galvanic action between the non-homogeneous portions of the iron or

steel plates. By placing the plates in connection with slabs of zinc, a galvanic action is set up between the iron and the zinc, which destroys the latter and leaves the former untouched.

In the British Navy zinc slabs 12 by 6 inches and $\frac{1}{2}$ inch thick are attached to the boiler stays, there being one slab to every 20 horsepower. These are eaten up and renewed about every 60 to 90 days. The zinc is reported to perform its duty very effectively.

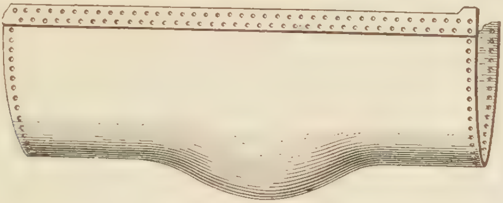
1873. External corrosion frequently attacks stationary boilers, particularly those set in brickwork. The causes of external corrosion are dampness, exposure to weather, leakage from joints, moisture arising from the waste pipes or blow-out. When leakage occurs in a joint which is hidden by the brickwork setting, the plates may be corroded very seriously without being discovered.

External corrosion should be prevented by keeping the boiler shell free from moisture, and by repairing all leaks as soon as they appear. Joints and seams should be in position where they may be inspected for leaks.

1874. Leakage at the seams may be caused by delivering the cold feed-water on to the hot plates; another cause is the practice of emptying the boiler when hot and then filling it with cold water. The leakage in both cases may be traced to the sudden contraction of the plates due to the sudden cooling. In any case abrupt changes in the temperature of the shell should be avoided. The rush of cold air into the furnace of an externally fired boiler when the door is opened is a fruitful source of leakage and fracture. For this reason the shell should be constructed, if possible, so that none of the seams are in contact with the fire.

1875. Overheating may be caused by low water or by incrustation. When the plate is covered by a heavy scale, the heat is not carried away by the water fast enough to prevent a rise of temperature, the plate becomes red hot and soft, and yields to the steam pressure, forming a pocket *A* as shown in Fig. 582.

If the pocket is not discovered and repaired, it stretches until finally the material becomes too thin to withstand the steam pressure; the pocket bursts, and an explosion follows.



A

FIG. 582.

The vegetable or animal oils carried into the boiler from the condenser are particularly liable to cause the formation of pockets.

INSPECTION AND TESTING.

1876. The condition of a boiler as regards safety can be determined only by careful inspection. Insured boilers are periodically inspected by experienced inspectors in the employ of the insurance company. The inspector notes the condition of the plates, whether or not they are corroded or incrusted, inspects the interior in search of broken stays or rivets or fractured joints. The condition of the plates is generally determined by tapping the plates with a light hammer; any weakness will immediately reveal itself to the skilled inspector, who is able to judge the thickness and soundness of the plate by the sound of the blow and the rebound of the hammer. When the thickness is a matter of doubt, a small hole may be drilled through the plate, and afterward plugged up.

1877. Boilers subject to government inspection are submitted to the hydrostatic test. The boiler is filled with water, a pump is applied and more water is forced in, until the pressure exceeds that which the boilers are expected to carry. If the boiler stands the water pressure without fracturing or developing leaks, it is assumed that it will carry the required steam pressure in safety.

In making the hydrostatic test the pressure must be applied very slowly and carefully, and the gauge watched

for any drop of pressure which would denote a yielding of some part of the boiler. New boilers are tested by hydrostatic pressure to reveal leaky joints or rivets. When the seams or rivets are not tight, water trickles out in drops or spins out in a stream. Such places are marked with chalk and afterwards re-calked. Boilers are usually tested hydrostatically to $1\frac{1}{2}$ times the pressure they are to carry.

It is objected against the hydrostatic test that there is danger of straining the plates beyond the elastic limits, and thereby a boiler may be permanently injured which would have been safe at the working steam pressure. The insurance companies in most cases depend upon the hammer test, but use the hydrostatic test for new boilers, old boilers extensively repaired, and all boilers which can not be examined thoroughly inside and outside.

A method of applying the hydrostatic test used by many engineers is to fill the boiler full of cold water and build a gentle fire in the furnace. As the temperature of the water rises it expands, and thus subjects the shell to pressure. It is urged in favor of this method that the pressure is raised steadily, and the boiler is not as liable to be injured as it is when subjected to sudden and jerky rises of pressure due to the working of a pump. The temperature of the water should in no case be made to rise above 212°, since, otherwise, if a rupture should take place, the pressure of the water would lower to that of the atmosphere, and the temperature of the water being above the boiling point at atmospheric pressure, a quantity of the water might suddenly flash into steam and cause an explosion.

The inspection of steam boilers should begin at the place where the plates are manufactured, and continue as long as the boiler is in use.

BOILER EXPLOSIONS.

1878. Boiler explosions are in nearly all cases due to one cause only—over-pressure of steam. Either the boiler is not strong enough to safely carry its working pres-

sure, or else the pressure has been allowed to rise above the usual point by the sticking or overloading of a safety valve, or some similar cause.

A boiler may be unfit to bear its working pressure for any of the following reasons: 1. Defective design. 2. Defects in workmanship or material. 3. Corrosion, and wear and tear in general. 4. Mismanagement in operation.

The following are faults in design which often lead to boiler explosions: The boiler is insufficiently stayed—the stays being too small or too few in number; the cutting away of the shell for the dome, manhole, and other mountings, without strengthening the edge of the plate around the hole; the boiler may be too rigidly fixed in its setting and thus fracture on account of unequal expansion; defective water circulation in a boiler may lead to excessive incrustation and consequent explosion; a poorly designed feed apparatus, or safety valve, often leads to explosion.

Defects of workmanship and material may include the choice of faulty material, containing blisters, lamination, etc.; the careless punching and shearing of the plates; burned and broken rivets; plates burned, or otherwise injured in flanging, bending, or welding; scoring of the plates along the joints by sharp calking tools; injury of plates by the reckless use of the drift pin. Old plates may be injured in patching them with new plates by the operation of removing the old rivets and putting in the new ones, and also by the greater expansion and contraction of the new plate when exposed to fire.

The strength of the shell may be weakened by corrosion, pitting, and grooving. In some boiler explosions, the plates have been found wasted to little more than the thickness of paper.

Fractures which ultimately end in explosion may be produced by letting the cold feed-water come directly into contact with the hot plates. It has already been remarked that the feed should be introduced into the coolest part of the boiler.

Vertical boilers hold the first place in the list of those liable to explosion. The ends of the tubes and the crown

sheet are very liable to corrode; the crown sheet bulges downwards, and the reaction of the escaping steam may throw the boiler high in the air. Again, explosion may be the result of the collapse of the upper ends of the tubes—an event which may occur when the tubes pass up through the steam space (see Fig. 498).

It has been previously explained how incrustation or the presence of grease may lead to the formation of pockets. If the pocket is not removed and replaced by a patch, rupture and explosion are liable to result.

The pressure of the steam may be allowed to rise above the normal blowing-off pressure by neglect or mismanagement. The safety valve may be neglected and allowed to stick fast to its seat. In one case a safety valve recovered from a boiler explosion was found to be corroded to such an extent that a pressure of $1\frac{1}{2}$ tons was required to start it from its seat. Again, the safety valve may be temporarily shut off from the boiler by a stop-valve; numerous very disastrous explosions have been due to this cause. It cannot be too strongly emphasized that *a stop-valve should never be placed between the safety valve and boiler.* Over-pressure and consequent explosion may be caused also by the practice of over-weighting the safety valve.

Low water is the cause ascribed to many explosions. It was formerly the custom to consider nearly every boiler explosion as due to shortness of water. It is now known, however, that externally fired boilers rarely explode on account of low water, though internally fired boilers may do so. In the case of boilers of the Lancashire type, a shortness of water leaves the furnace flue uncovered, it becomes overheated and is liable to collapse; the same is true in regard to the fire-boxes of the locomotive type. On the other hand, the tubes of a return tubular boiler may stand a large amount of overheating before giving way, and it is next to impossible to burst a plain cylindrical boiler by low water, as long as it contains any water at all. Low water may lead to explosion, however, in the following manner: The uncovered plates become red hot, and upon

being suddenly covered with fresh cold feed-water, may suddenly contract to such an extent as to produce rupture.

The superheating of the water has been supposed to have produced explosions in rare instances. It has been observed that water from which the air has been expelled may be heated 10° or 20° above its normal boiling point without any signs of boiling. Upon being agitated, as by the introduction of fresh feed-water, the volume of superheated water flashed suddenly into steam might, under favorable conditions, produce a pressure great enough to cause an explosion.

A weak boiler may possibly be exploded by the sudden opening or closing of a throttle valve. That such an explosion is possible is shown by the following experiment, made by United States Inspectors: A cylindrical boiler was tested and withstood a steam pressure of 300 pounds without injury. It was then filled again, and the steam pressure was run up gradually, the discharge valve being opened at intervals and the fall in pressure noted. When the valve was suddenly opened at a pressure of 235 pounds, the boiler was torn into fragments, the iron being twisted and torn and thrown in all directions. The sudden rush of steam from the boiler into the discharge pipe reduces the pressure in the boiler very rapidly; the reduction of pressure causes a sudden formation of a great quantity of steam within the water, and the heavy mass of water is thrown towards the opening with great violence, strikes the portions of the boiler near the opening, and breaks it open.

Explosions from this cause are probably rare; still, it is well to use caution in opening or closing stop-valves or safety valves.

1879. The destructive nature of a steam-boiler explosion is due to the enormous amount of energy stored in the steam and heated water. Professor Thurston calculates that a cubic foot of water heated at constant volume until its pressure is 60 to 70 lb. per sq. in. has about the same energy as a pound of gunpowder, and that the energy stored

in a plain cylindrical boiler at 100 pounds pressure is sufficient to project it to a height of over $3\frac{1}{2}$ miles.

If a boiler fractures while undergoing the hydrostatic test, the water escapes through the rent in the plate and no explosion takes place, because the cold water has little or no stored energy. But when a boiler filled with steam and water at high temperature fractures, a violent explosion generally follows. The steam escaping through the opening diminishes the pressure, and, consequently, a new body of steam is formed from the water, which, by escaping, lowers the pressure still more, allowing the formation of another new body of steam at a lower pressure, and this operation is continued until the pressure reaches that of the atmosphere. This formation of successive large bodies of steam, occurring, as it does, almost instantly, can not fail to produce a disastrous explosion. Generally speaking, the larger the body of the contained water, the more disastrous is the result. The safety of water tubes and sectional boilers is thus accounted for by the fact that they consist of numerous parts or sections, each containing a relatively small body of water. The bursting of one of these sections is unusual on account of its small diameter, and should it occur, no disastrous explosion would be likely to follow, on account of the small quantity of contained water.

Boiler explosions may be prevented by the use of a properly designed and well-made boiler, which has been correctly set and is under careful management. The boiler must be regularly inspected, repaired when necessary, and removed before becoming so worn out as to be dangerous.

Where boilers are to be placed in buildings, safety is the first object to be sought, and some form of sectional or water-tube boiler should be used. Where considerations of cost overrule those of safety, the tubular boiler is generally adopted.

MANAGEMENT AND CARE OF BOILERS.

1880. The following are the rules and directions generally given for the guidance of firemen and attendants in the performance of their duties:

1. **Firing.**—Keep the fire of uniform thickness, and allow no airholes in the bed of fuel. Fire evenly and regularly, and not too much at a time. Keep the fire free from ashes and clinkers, and as clean at the corners and edges as at the center. Keep the ash pit clear. Do not clean the fires oftener than necessary.

2. **Water-Level.**—The first duty of the fireman upon going to work should be to examine the water-level. The gauge-cocks should be tried; the gauge glass is not always reliable. In a battery of boilers, the gauge-cocks on *each* boiler should be tried. Some serious explosions have resulted from the fact that the fireman only consulted the water-level in the first boiler, and took it for granted that the level in the other boilers was the same.

3. **Low Water.**—If the water is discovered to be low, quickly cover the fire with ashes, or, if they are not convenient, with fresh coal. Do not turn on the feed, and do not tamper with the safety valve or any other steam outlet. The fire may be drawn as soon as it can be done without increasing the heat.

4. **Foaming or Priming.**—In case of foaming close the throttle valve of the engine or the stop-valve of the supply pipe, and keep it closed long enough to show the true water-level. The foaming can generally be stopped by blowing off and feeding fresh water. In cases of violent foaming, due to dirty water or to a change from fresh to salt water, check the draft and cover the fire with fresh coal.

5. **Leaks.**—When leaks are discovered they should be repaired at once.

6. **Blowing Off.**—When blowing off, the steam pressure should be not over 20 pounds. The boiler should be emptied at least once every two weeks and filled up afresh. Once every week would be better. If the water is muddy, blow out 6 or 8 inches every day. A surface blow-out should be opened for a short time at frequent intervals. Examine the blow-out tap and check-valves every time the boiler is filled; a leakage from either may lead to serious results.

7. **Filling Up.**—Allow the boiler to become cool before pumping in cold water; the practice of filling a hot boiler with cold water causes leaks and fractures, and sometimes explosions.

8. **Safety Valves.**—Raise the valves from their seats cautiously and frequently. Do not allow the valve to be overloaded.

9. **Pressure Gauge.**—The pressure gauge should stand at 0 when the steam pressure is off, and it should indicate the blowing-off pressure when the safety valve is in action. If the gauge does not do this, it should be compared with a standard gauge, and, if wrong, corrected.

10. **Gauge-cocks** and **gauge glasses** should be kept clean and in constant use. The water gauge should be blown out frequently and the glasses and passages to gauges kept clean. An obstructed gauge sometimes shows a false water-level. The Manchester Boiler Association attributes more accidents to this cause than to all others combined.

11. **Feed-Pump and Injector.**—Both pump and injector should be of ample size, and whichever is used should be made to work as uniformly and continuously as possible. It is best to have two independent means of feeding the boiler. Check-valves should be frequently examined.

12. **Removal of Sediment and Incrustation.**—Scale and sediment should be frequently removed. In tubular boilers, particularly, the handhole or manhole should be frequently opened and the sediment removed from the portions of the plate over the furnace. Care should be taken to keep the boiler as free as possible from incrustation.

13. **Cleaning.**—All heating surfaces should be kept free from soot and dirt. Tubes should be often cleaned.

14. **Exterior of Boiler.**—Care should be taken that no water comes in contact with the exterior of the boiler, either from leaky joints or other sources. Avoid dampness in the setting or in the covering of the boiler. Dampness leads to external corrosion.

15. **Blisters and Cracks.**—A blister should be at once examined, and trimmed or patched. If a plate is badly cracked it should be renewed.

16. **Fusible plugs** should be examined when the boiler is cleaned, and scraped clean on both sides; otherwise they are liable to prove worthless.

17. **Air Leaks.**—See that the furnace, combustion chamber, and smoke flue are tight. The admission of air through the brickwork of the setting is sometimes a source of considerable loss.

18. **Galvanic Action.**—Examine the parts of the boiler where brass or copper and iron come in contact in the presence of water. Galvanic action may produce corrosion under such circumstances; if such be the case, the corrosion may be prevented by placing pieces of zinc in the boiler.

19. **Rapid Firing.**—Steam should be raised slowly in boilers with thick seams, or seams exposed to the fire. Otherwise overheating may occur.

20. **Cleanliness.**—The boiler room, boiler, and mountings should be kept clean and in good order.

21. **Unused boilers** may be kept in good condition by filling them full of water in which a quantity of common washing soda has been placed. Another method is to empty the boiler, dry it thoroughly, place trays of quicklime in the bottom, and seal it as nearly air-tight as possible. The latter method is often used for marine boilers.

BOILER SETTING.

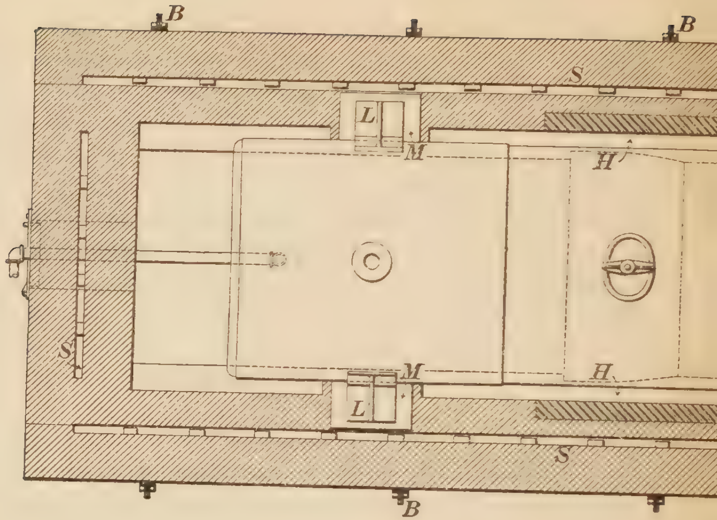
1881. In boiler setting three things are to be attained. 1. A stable support for the boiler shell. 2. Properly arranged space for furnace and ash pit. 3. A protective covering for the boiler which shall, as far as possible, prevent loss of heat by radiation.

1882. Externally fired boilers may be supported by cast-iron lugs riveted to the shell and resting on the side walls, or they may be suspended from overhead girders by

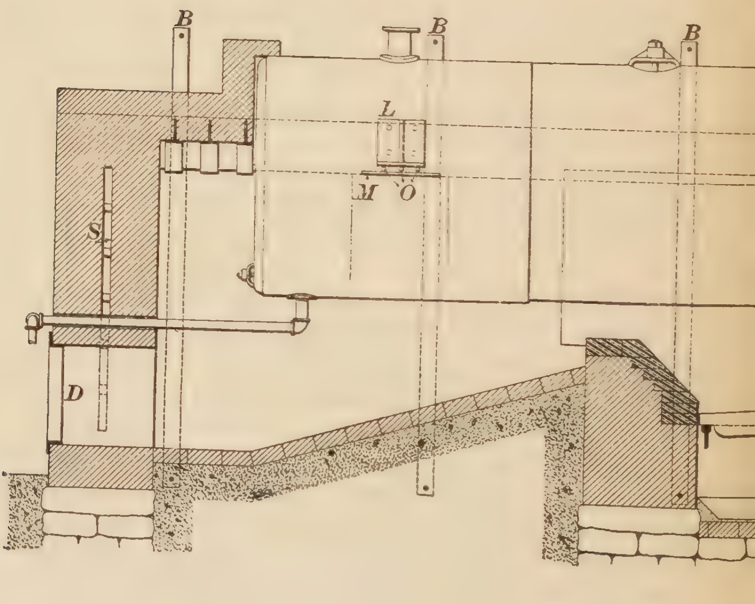
means of hooks or rings. The former method is usually adopted for the comparatively short return tubular boiler, while the latter is generally used for the long plain cylindrical and flue boilers. When very long cylindrical boilers are suspended at two points only, the excessive weight between the supports throws a great strain on the lower plates in the middle of the boiler; when a center support is added, the condition of things is still worse, because the lower part of the shell expands more than the upper, causing the shell to sag in the middle, and thus throwing all the weight on the center support. Numerous cases have occurred where the center support has given way under the strain and the shell ruptured by the shock. It is, therefore, important in supporting these long shells to arrange the supports so that each shall bear its proper proportion of the load, and at the same time allow the boiler to expand freely under all conditions of temperature.

1883. The setting of a 60-inch return tubular boiler, as designed by the Hartford Boiler Insurance Company, is shown in Fig. 583. The foundation is made by digging down three or four feet and laying in heavy stonework; on top of this the brickwork is laid. The side and rear walls are double, with a 2-inch air space between the inner and outer parts. The inside wall *I* next to the furnace is faced with firebrick, as is also the bridge and all portions in direct contact with the flames.

The boiler is supported by cast-iron lugs *L* riveted to the shell. These lugs rest upon iron plates *M* placed upon the top of the side walls. The front lugs rest directly upon the plates, but the back lugs rest upon rollers *O* of 1-inch round iron. The boiler is thus free to expand and contract. The rear wall is 24 inches from the rear head of the boiler to allow the gases opportunity to enter the tubes. Above the tubes, however, the wall is built in to meet the head and forms a roof for the chamber. The rear wall is provided with a door *D* for the removal of the dirt and soot that collects back of the bridge, and provides a means of inspection,

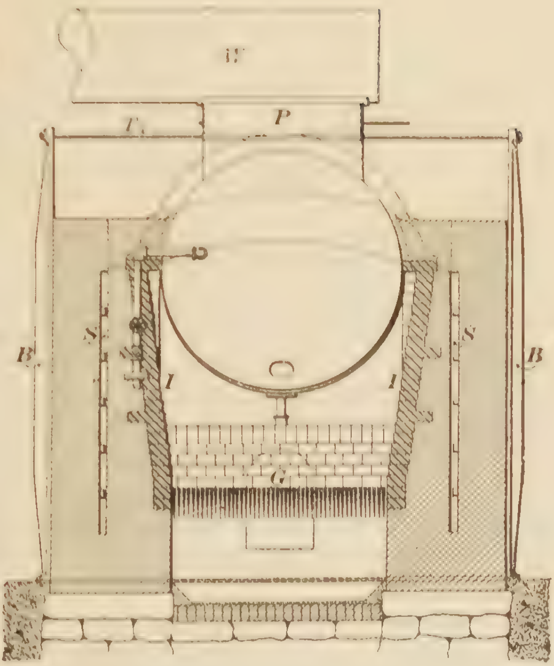


B
B
S
L
M
H
S
M
H
L
S
B



B
B
B
L
M
O
S
D


L
M
P
M
L
B
W
E
P
L
M


W
P
T
B
S
I
I
S
B
G

The grate *G* is placed 24 inches below the shell; this is a sufficient distance for anthracite coal, but for bituminous it might better be 28 or 30 inches. The grate has a fall of 3 inches from front to rear, so that the fuel bed is thicker near the back end of the fire; this is believed to lead to more even combustion, since the air has naturally a greater tendency to pass through the fire nearest the bridge, and upon meeting a thick bed of fuel its passage is somewhat retarded.

The end of the boiler to which the blow-off pipe is attached should be set about 1 inch lower than the other end; this aids in blowing out sediment and in draining the boiler through the blow-off pipe.

The brickwork is closed into contact with the shell at the level of the center of the upper row of tubes; this prevents the gases from coming in contact with the plates above the water-line. Some boiler-makers prefer to make a brickwork arch over the top of the boiler and allow the gases to pass back to the rear through the flue thus formed. The practice is risky, as it may lead to the overheating of the upper plates. A safe rule is, "Never expose to fire or to gases of combustion any part of the shell not completely covered by water."

The brickwork is strengthened by buckstaves *B*, held together by tie-rods *T*. The buckstaves are best made of wrought-iron channel or angle irons. It will be noticed that in the present case the flue pipe *P* is rectangular, but the pipe *W* leading to the chimney is cylindrical. The air spaces *S* are for the purpose of securing a circulation of air through the walls and thus keeping them cool. Their utility is somewhat doubtful, and many of the best boiler-makers do not recommend them.

The settings of the various types of water-tube boilers have been clearly shown in the cuts accompanying these descriptions.

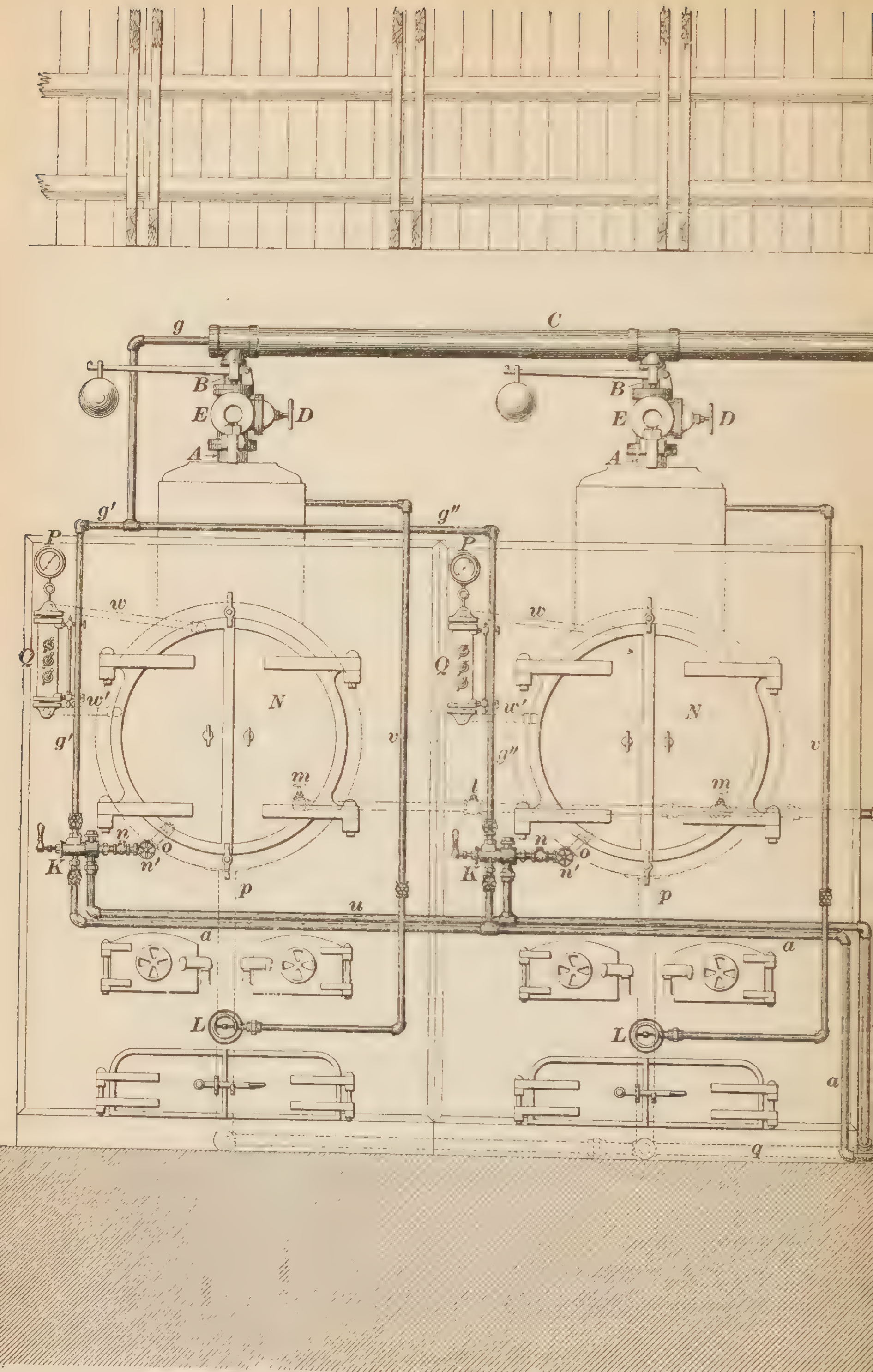
Internally fired boilers require a setting only for the purpose of forming a support and covering. Boilers of the Cornish and Lancashire type are set in brickwork, as shown in Figs. 492 and 494. The boiler rests on two narrow ridges

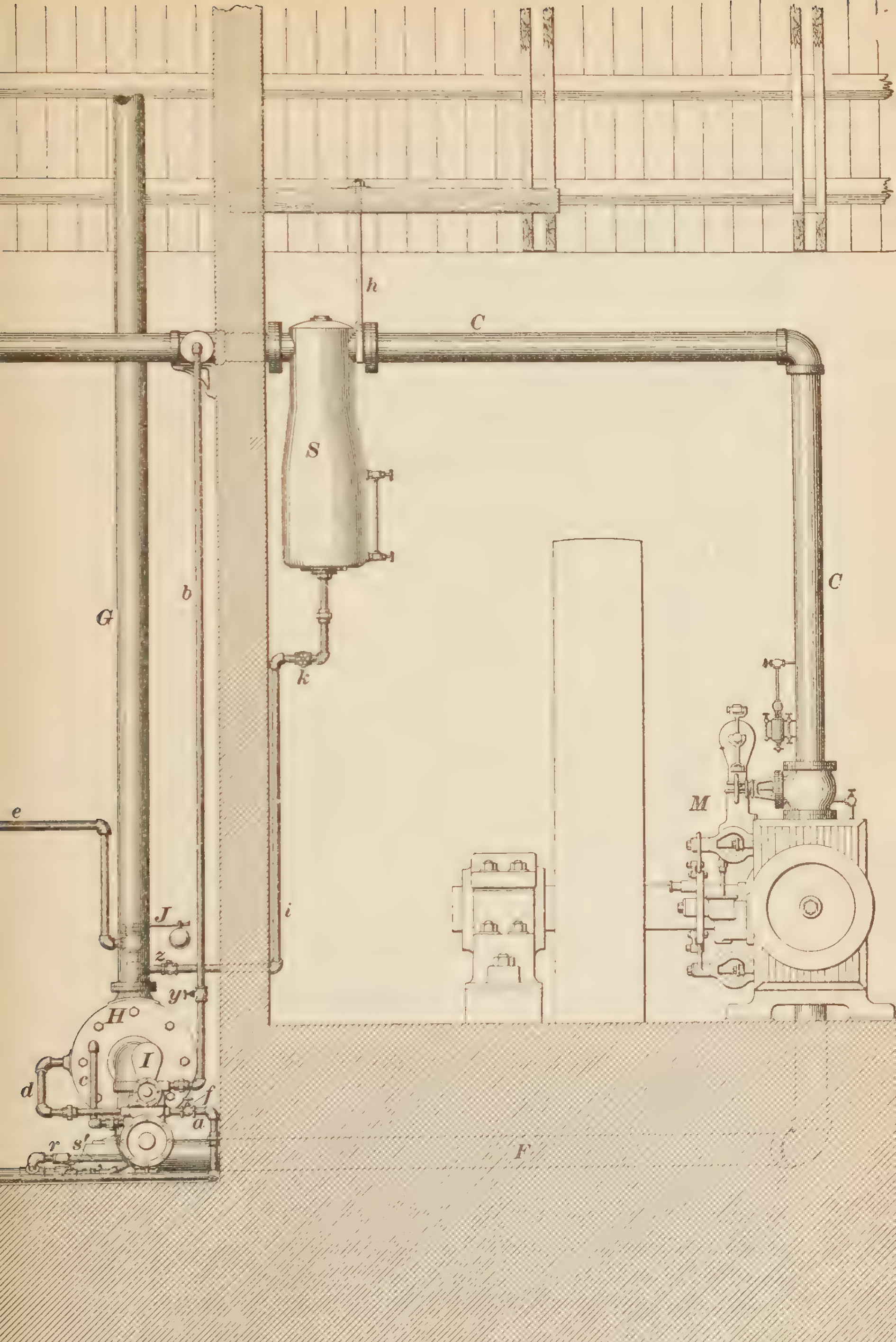
of firebrick, which form the boundary between the bottom and side flues. The main point to be regarded in setting this type of boilers is to avoid having an excessive brickwork surface in contact with the boiler shell; the brickwork is liable to collect moisture and lead to external corrosion.

Plain cylindrical and flue boilers are set in about the same manner as the return tubular. Sometimes, however, when the shells are extremely long, two or even more bridges are placed beneath the shell to keep the heated gases in contact with the boiler.

Vertical and locomotive boilers are self-contained, and require no setting. The vertical boiler is supported by the cast-iron base which forms the ash pit. Fire-box boilers, when stationary, are supported on cast-iron saddles and skids. It is not customary to provide vertical boilers and stationary fire-box boilers with any protective covering.

1884. Boiler and Pipe Coverings.—The tops of boilers and other portions of the surface not in contact with the furnace gases should be covered with some non-conducting substance to prevent the radiation of heat. The same is true of the steam pipes leading from the boiler to the engine. A common and effective covering for the tops of boilers is ashes or loam. Where several boilers are set in a row, the ashes or loam are filled into the space between them until the surface is level. This form of covering is open to the objection that the ashes tend to gather moisture and hasten external corrosion, especially if the shell leaks under the covering. Another way of covering the exposed surfaces is to plaster them with a mortar of $\frac{1}{3}$ plaster-of-paris and $\frac{2}{3}$ sawdust and afterwards cover them with a coating of asbestos, hair felt, or mineral wool, which may be tied on with wire. The whole is then covered with roofing paper. The following covering for steam pipes is recommended: "First wrap the pipe in asbestos paper; then lay strips of wood lengthwise along the pipe, from 6 to 12 in number, according to the size of the pipe, and bind them with wire or cord. Around this framework wrap roofing paper and fasten it with paste or


g
C
B
E
D
A
g'
g''
P
w
Q
w'
N
v
m
l
K
n
n'
o
p
u
a
L
q



h
C
S
C
G
b
k
M
e
J
i
z
y
H
I
c
d
f
a
r
s'
F

twine. If exposed to weather, use tarred paper or paint the exterior." Where flanges occur, space may be left to give access to the bolts and afterwards filled up with hair felt.

Coverings are now manufactured by some concerns to fit the various sizes of steam pipe; they may be easily fastened to the pipe, and as easily removed. They are generally made of magnesia, or asbestos, or a combination of the two, and appear to serve their purpose effectively.

1885. The following table showing the relative values of non-conducting substances is due to Mr. C. E. Emery:

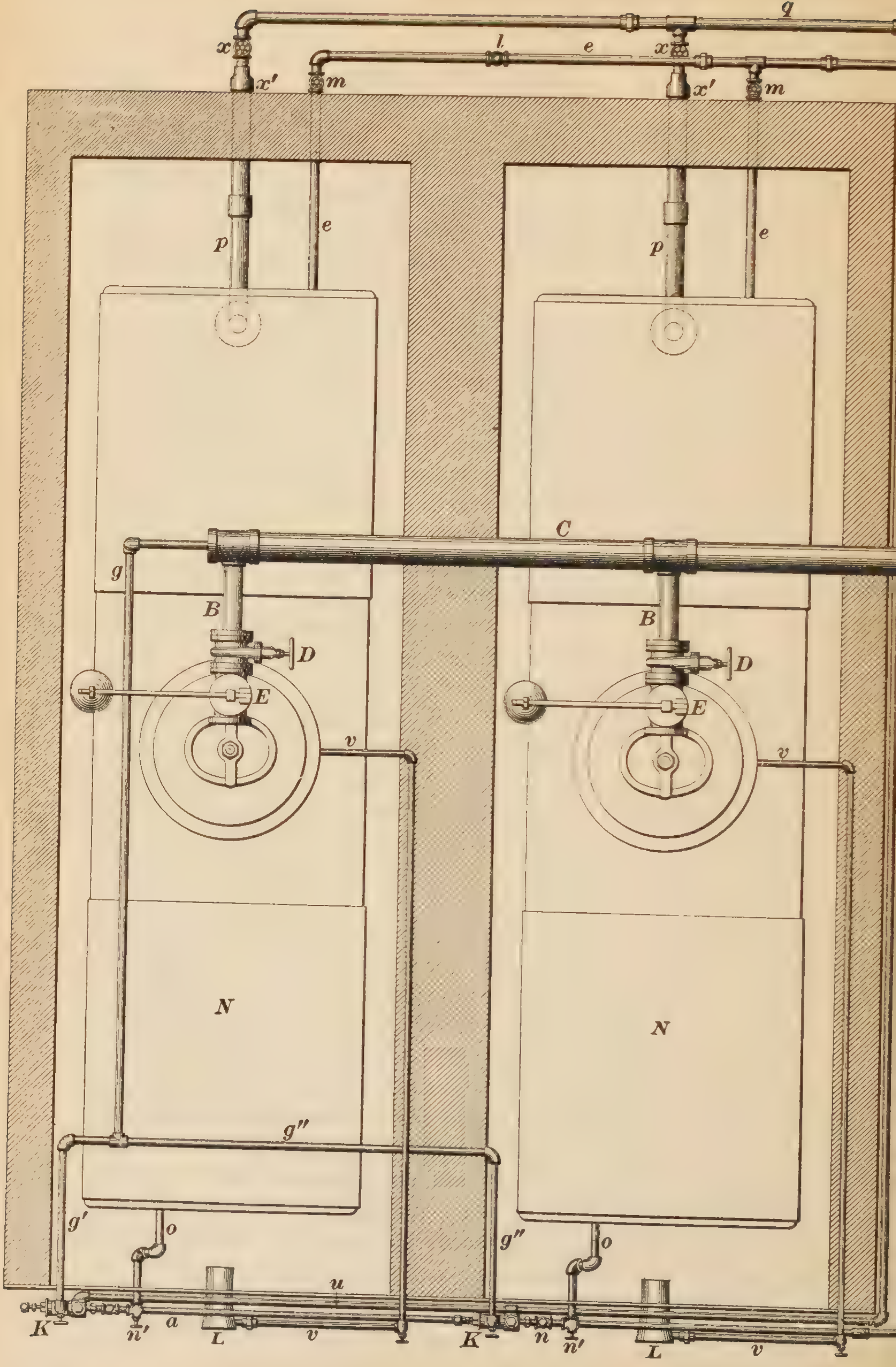
Material.	Value.
Hair felt	100
Mineral wool No. 2	83.2
" " " 2 and tar	71.5
Sawdust	68
Mineral wool No. 1	67.6
Charcoal	63.2
Pine wood across grain	55.3
Loam	55
Slacked lime	48
Gas-house carbon	47
Asbestos	36.3
Coal ashes	34.5
Coke in lumps	27.7
Air space 2 inches deep	13.6

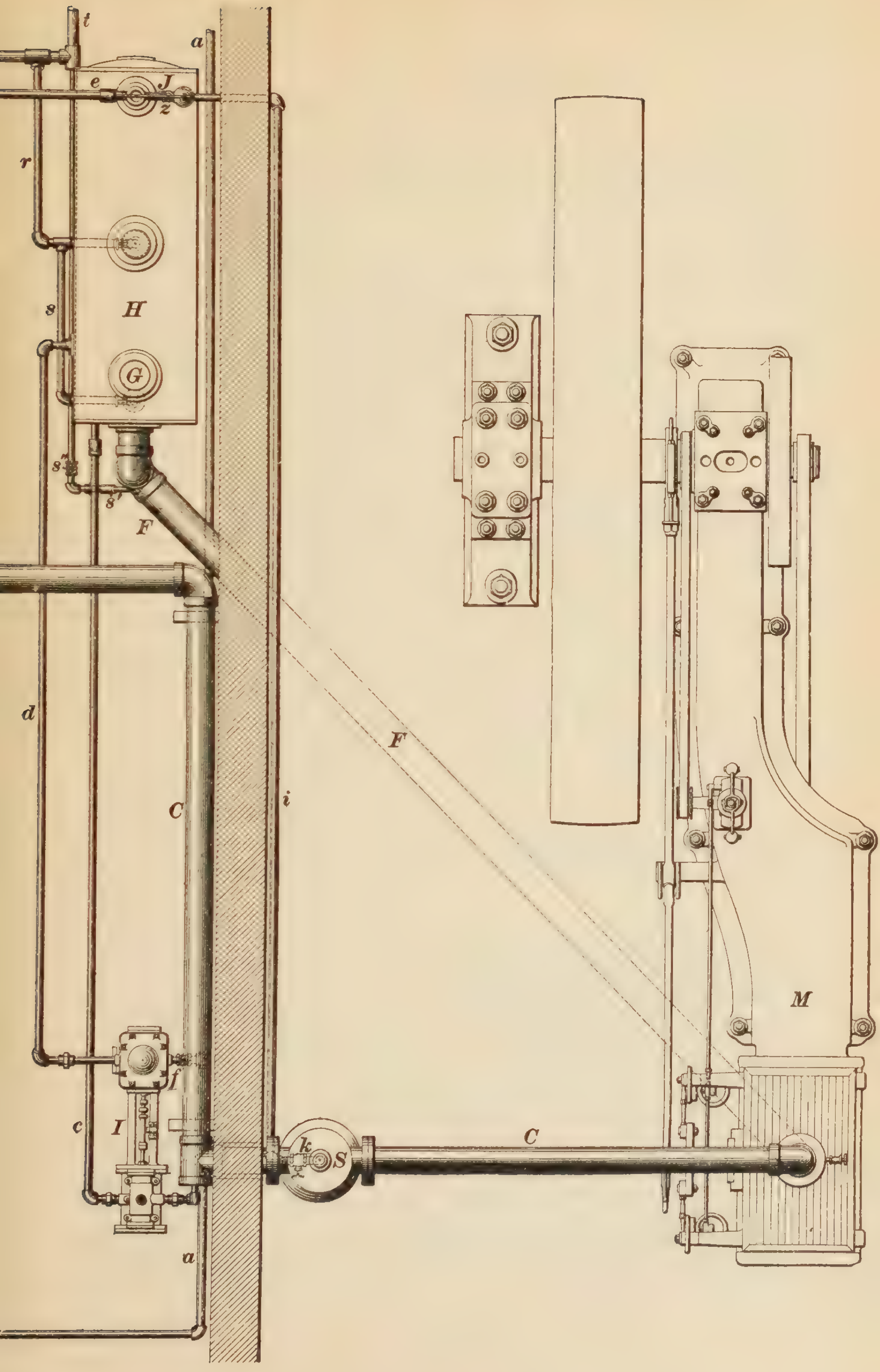
1886. The **installation** of a complete steam plant includes the setting of the boiler or boilers, the arrangement of the various lines of piping, and the location and arrangement of the various accessories, such as feed-water heaters, purifiers, separators, economizers, feed-pumps, injectors, etc. An elaborate plant may be fitted with economizers, mechanical stokers, coal-conveyers and ash-conveyers, purifiers, and other labor-saving and fuel-saving devices. On the other hand, the plant often consists simply of boilers, chimney, and feed-pump.

A good arrangement of a modern boiler plant is shown in Figs. 584 and 585. Fig. 584 is an elevation, and Fig. 585 a plan of the arrangement, which consists of two return tubular boilers *N*, *N*. In the following description the letters refer to both Fig. 584 and Fig. 585.

Suppose that the boilers have been partly filled with water, the fire started, and that the steam gauge *P* registers the desired pressure. The valves *D*, *D* are then opened and the steam is conveyed to the engine *M* through the short vertical pipes *A*, *A*, the short branch pipes *B*, *B*, and the main steam pipe *C*. It will be noticed that the safety valves *E*, *E* are attached to the upper ends of the pipes *A*, *A*, and that the valves *D*, *D* are situated between the safety valves and the main steam pipe, so as to prevent any liability of an explosion which might occur through carelessness in leaving the valves *D*, *D* closed when firing up if they were between the safety valve and the boiler.

H is a feed-water heater, the water being heated by the exhaust steam from the engine, which is conveyed to the heater by the exhaust pipe *F*. The water in the boiler is replenished by means of the feed-pump *I*. The pump is connected to a city reservoir, river, or other source of supply by the supply pipe *a*, and discharges through the delivery pipe *d* into the heater *H*, and from thence, after being heated, the water is forced by the continued working of the pump into the boilers through pipe *e* and its branches; *l*, *l* are check-valves to prevent the return of the water when the pump reverses its stroke. The valves *m*, *m* are for the purpose of shutting off the water from one of the boilers, if so desired. The steam for working the pump is received from the main steam pipe *C* through the small pipe *b*. The throttle valve *y* is used for starting or stopping the pump. The exhaust steam from the pump is conveyed through the pipe *c* to the heater, and mixes with the exhaust from the engine. The exhaust steam leaves the heater through the pipe *G* and is conveyed to some point beyond the boiler house, where it is discharged into the atmosphere. It will be noticed that a safety valve *J* is situated on the pipe *e*, or


q
x
l
e
x
x'
m
x'
m
e
p
e
p
C
g
B
B
D
D
E
E
v
v
N
N
g''
g'
o
g''
o
u
K
n'
a
L
v
K
n
n'
L
v



t
a
e
J
z
r
H
s
G
s'
s'
F
d
F
C
i
M
f
c
I
k
S
C
a

rather on the short branch pipe between it and the feed-water heater; this is necessary in order to prevent any over-pressure in the heater.

The blow-off pipes p, p are connected to the pipe q, which in turn is connected to the pipe t. The valves are shown at x, x. The pipes p are larger than q, for the reason that they are more likely to become choked with sediment than q. Should they become choked, they may be readily disconnected at x' from the pipe q and the sediment removed. The pipe r is the blow-off for the heater; it connects with the main blow-off pipe q. Since the exhaust steam condenses more or less in passing through the heater, it is necessary to provide some means for getting rid of the condensed portion; this is accomplished by connecting the tubes to the blow-off pipe r by means of the pipe s. A small drain pipe s' is fitted to the exhaust pipe F and connects with the blow-off pipe t. The globe valve s'' should be opened before the engine is started, so as to clear the exhaust pipe of water that may have accumulated in it. This valve should be closed again after the exhaust pipe is thoroughly warmed up and cleared of all the water. When the boilers are being blown off, a globe valve on the pipe s (not shown in the illustrations) is closed.

L and L are Argand blowers for producing a force draft, the steam required being obtained from the dome by the pipe v.

In case the pump should be out of order, the boilers can be fed by the injectors K, K. The steam for working the injectors is obtained from the main steam pipe C and conducted to them by the small pipe g and its branches g' and g''. The injectors are fed by the pipe a, a continuation of the pump feed-pipe a, and delivers the water into the boiler through the feed-pipes o, o. Before starting the injectors, the valve f should be closed, so as to shut off the water supply from the pump. The pipe u conducts the overflow water from the injectors to the blow-off pipe q. The check-valves n, n prevent the water from escaping from the boiler through the injectors after they have stopped working. The globe

valves n', n' are additional safeguards; they are for the purpose of preventing the boiler from emptying itself after a shut-down in case an obstruction should prevent the check-valves from closing.

The steam gauges *P*, *P* and the water-gauge cocks and glasses are connected to cast-iron columns *Q*, *Q*. The tops of these columns are connected to the steam space in the boilers by the pipes *w*, *w*, and the bottoms to the water space by the pipes w', w'.

S is a separator for removing the entrained water from the steam; its action is described in Art. **1895.** As will be seen, it is attached to the main steam pipe *C*, and also supported by the rod *h*, which is attached to the beam overhead. The water thus removed flows down by gravity through the pipe *i* into the feed-pipe *e*, just below the safety valve *J* on the feed-water heater. As its temperature is fully 212°, it is not necessary that it should pass through the heater. The bottom of the separator should be at least two feet or more above the highest water-level in the boiler, since if both were at the same level the pump would force water into the separator and thus destroy its action. The difference of levels between the bottom of the separator and the water-level in the boiler constitutes the head which induces the flow. The pipe *i* is fitted with the globe valve *k* and the check-valve *z*.

1887. Steam Pipes.—The piping of a boiler plant is a point that must be carefully attended to. The main steam pipe, or "log," as it is sometimes called, is generally made of cast-iron sections flanged and bolted together. Where several boilers discharge into the same log, some provision must be made for expansion and contraction. If the long line of cast-iron pipe is connected directly with the boiler shell, the expansion due to the entrance of hot steam will twist and wrench the pipe and cause leaky joints, if nothing more serious. The most approved method of connecting up the main pipe with the boilers is shown in Fig. 586, which is a side view of the dome and pipe connections of Fig. 584.

From the dome, or from the shell, if there is no dome, the short length of pipe A rises vertically and connects with a horizontal branch B, which is joined to the main pipe C by the short vertical pipe A'. The main steam pipe is thus allowed to expand or contract freely. The only effect of the expansion is to slightly turn the pipe B on A as a pivot. In designing the main line of pipe, the section between boilers 1 and 2 need only be large enough to carry the steam from boiler number 1. The section between boilers 2 and 3

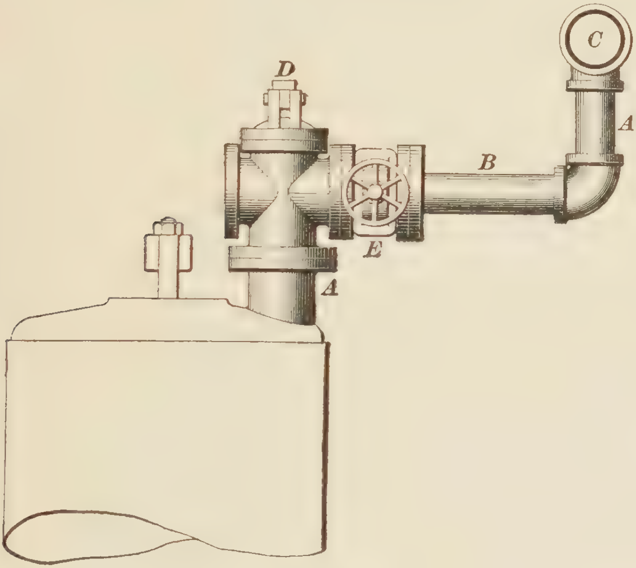


FIG. 586.

should have, then, twice the sectional area of the first section, the portion between boilers 3 and 4 should have 3 times the sectional area of the first section, and so on. Thus, if the diameter of pipe B is 5 inches, the diameter of the first section of the main pipe will also be 5 inches; the diameter of the next section should be $5\sqrt{2} = 7$ inches; of the third section $5\sqrt{3} = 8\frac{1}{2}$ inches, nearly; of the fourth section $5\sqrt{4} = 10$ inches.

1888. Another way of providing for the expansion of the main pipe is to use a length of curved copper pipe *C* in place of *B*, Fig. 586. (See Fig. 587.) The curved form of the copper section enables it to take up all the expansion or contraction of the log, and also its own expansion or con-

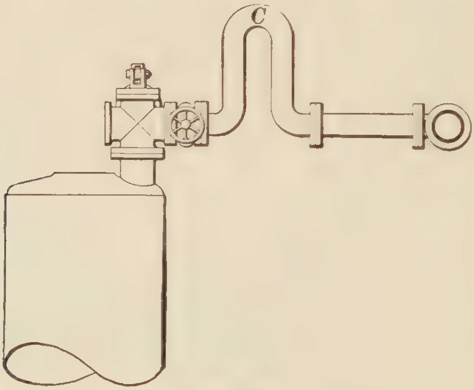


FIG. 587.

traction. The ends of the copper pipe are brazed to brass flanges, which are bolted to the flanges of the cast-iron sections.

1889. In some of the most modern large plants the steam pipes are made of sections of steel pipe expanded into or riveted to steel flanges. The ordinary cast iron **L** 's are replaced by sections of steel pipe, bent to a long radius, and, when long lengths of pipe occur, expansion and contraction are provided for by means of **U** shaped sections of this pipe.

Steel pipe has the advantage of being much stronger, more flexible, and reliable than cast-iron pipe, properties that are especially desirable for the high pressures common where compound engines are used.

1890. The diameter of a steam pipe can be calculated on the assumption that the velocity of the steam is not to exceed 6,000 feet per minute. Suppose, for example, it is required to find the diameter of a steam pipe which is to convey 3,000 pounds of steam per hour, the pressure of the

steam being say 85 pounds. From the steam table, the volume of a pound of steam at this pressure is 4.35 cubic feet, nearly. Hence, the volume discharged per minute is $\frac{3{,}000 \times 4.35}{60}$ cubic feet, and the required area of pipe is $\frac{3{,}000 \times 4.35}{6{,}000 \times 60}$ square feet $= \frac{3{,}000 \times 4.35 \times 144}{6{,}000 \times 60} = 5.22$ square inches, or the *minimum* diameter is $2\frac{5}{8}$ inches. The diameter to be adopted is something above the minimum; probably in this case 4 or 5 inches would be used.

1891. The rules for calculating the thickness of cast-iron or steel pipes for various pressures have already been given in Art. **1365.** According to Unwin, the thickness of copper steam pipes is

$$t = 0.0001\, p\, d + \tfrac{1}{8}, \qquad \textbf{(206.)}$$

where t is the thickness, p the gauge steam pressure, and d the diameter of pipe. The thickness of the brass flange of the copper pipe is 3 times the thickness of the pipe. Thus, a 4-inch copper pipe under a pressure of 60 pounds would be $(.0001 \times 60 \times 4) + \frac{1}{8} = .149$ inch thick.

1892. The sections of cast-iron or steel pipe are bolted together at the flanges. The joint is made tight by facing the flanges and interposing between them a ring of india-rubber or gutta-percha. Sometimes a corrugated copper plate is placed between the flanges, and again a groove is cut in them and a copper wire placed in it.

For steam pipes of small diameter the ordinary wrought-iron gas pipes and fittings are used.

1893. Steam pipes should be so arranged that no pockets or angles are formed in which water may collect. Where this can not be avoided, the bend or angle should be provided with a drain cock or trap. The presence of water in a steam pipe is the cause of the so-called "water hammer" which is so often heard in steam-heating plants. Prof. Thurston has experimentally shown that the pressure produced by "water hammer" may be ten times that which

the pipe was expected to sustain in its regular work. In some cases the "water hammer" has caused boiler explosions by bursting the steam pipes.

STEAM APPLIANCES.

1894. A **separator** is an apparatus designed to remove the entrained water, or the oil, dirt, and other impurities, from a current of steam flowing through a pipe. When the separator is intended to free the steam from water, simply, it is placed on the main pipe leading from the boiler to the engine, and as close as possible to the latter. When it is desired to remove the grease and dirt from the exhaust steam before condensing it and feeding it back to the boiler, the separator is placed in the exhaust pipe leading from the engine to the condenser.

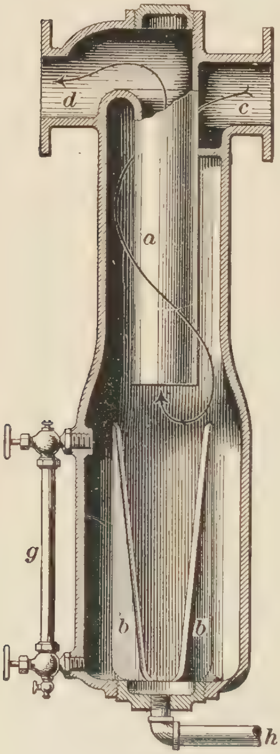


FIG. 588.

1895. The Stratton Separator is shown in Fig. 588. It consists of a chamber with a steam inlet and outlet, and containing a vertical pipe *a*. The steam enters by the inlet *c*, and is deflected by a curved partition, which gives it a spiral motion about the pipe *a*. The particles of steam are thrown off by centrifugal action and run down the walls to the bottom of the chamber. The steam passes up through the pipe *a* and out of the outlet *d* in a practically dry condition. The separator is provided with a drip pipe *h* for the removal of the water, and a gauge glass *g*. The wings *b*, *b* are four in number, and are for the purpose of destroying the centrifugal effect of the steam after it has reached

the bottom of the separator. They likewise offer additional surface for the water particles to adhere to. Were it not for these wings the steam would keep up its rotative motion while passing up the pipe *a*, and thus necessarily carry some of the entrained water along with it.

There are many other makes of separators, all, however, operating on practically the same principle. What is required of a separator is to check the velocity of the steam and change the direction of the current. The particles of water will continue in the original direction of the current by reason of their inertia, while the dry steam passes off in another direction.

1896. The **steam loop** is an appliance for returning the water of condensation from a steam pipe or separator to the boiler. The arrangement of the loop is shown in Fig. 589. It consists of a system of piping extending from the separator *c* back to the boiler. A vertical pipe *d*, called the *riser*, extends vertically upwards a suitable distance and empties into the pipe *e*, called the *horizontal*. Another vertical pipe *f*, called the *drop leg*, connects the horizontal with the boiler. The loop is provided with stop-valves and check-valve merely as a matter of convenience.

The action of the loop is as follows: The pressure at the separator is somewhat less than at the boiler; suppose, for example, that the pressure in the boiler is 75 pounds, and at the separator, 70 pounds. Then, the pressure of 70 pounds will extend back through the loop, and in the drop leg there will be a column of water about 10 or 11 feet high, due to the 5 pounds drop in pressure. That is, the 70 pounds pressure in the loop added to the pressure due to the column of water must just balance the boiler pressure. A portion of the steam in the horizontal condenses, reducing the pressure to say 69 pounds. The water column in the drop leg rises, of course, on account of the decrease of pressure, but the column of mixed steam, water, and spray also ascends in the riser to take the place of the condensed steam, and, being lighter than the water column in the water leg, rises much

higher. The riser then empties its contents into the horizontal, from which it passes into the water leg and into the

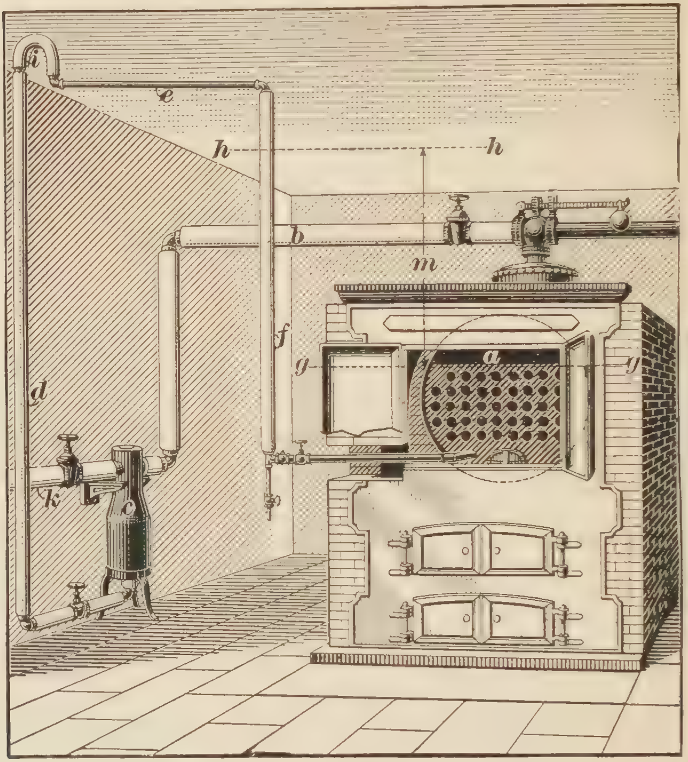


Fig. 589.

boiler. The loop thus carries all the water from the steam pipe and separator back into the boiler as fast as it appears.

1897. The **injector** has already been described in Art. **1076.** It is an instrument by means of which the energy of a jet of steam from the boiler is used in forcing a stream of water back into the boiler. The diagram, Fig. 590, will explain the action of the injector. Suppose A to represent the section of a boiler, and B a steam pipe leading from it and terminating in a nozzle which discharges into the conical

vessel *C*. The steam escapes from the boiler with great velocity, and as it passes through the cone *C* draws the air along with it, and thus creates a partial vacuum in the pipe *E*. The atmospheric pressure forces the water in the vessel *F* up into the nozzle *C*, through the pipe *E*, and there meeting the steam condenses it. Though condensed from steam to water the jet still has sufficient velocity to pass into the pipe *G*, and thence back again into the boiler. Not only has the jet sufficient energy to force itself into the boiler, but it also imparts enough energy to an additional mass of water to enable it also to enter the boiler. The energy of the jet is derived from the heat given up by the condensation of the

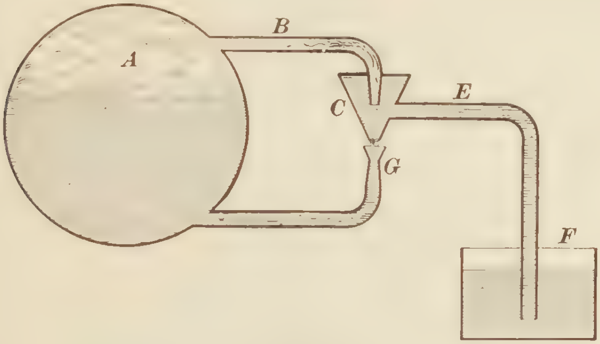


FIG. 590

steam. That this energy is sufficient to do the work required is evident when we consider that each pound of steam condensed gives up 966 B. T. U., and each B. T. U. is equivalent to 778 foot-pounds. In feeding boilers only part of the available energy is used in forcing the water into the boilers; some is wasted, and the remainder is expended in heating the feed-water.

1898. The Penberthy, one of the leading forms of injectors, is shown in Fig. 591. Steam enters from the boiler, passes through the connection *V* and nozzle *R* into the conical combining tube *S*. Here it meets the water, which enters through the pipe *P* (shown on the side in the cut, but really situated in the rear) and surrounds the nozzle *R*. At

first the mingled steam and water flows through the valve Q and out of the overflow pipe T; but the condensation of the steam soon causes a partial vacuum in the upper chamber, and the atmospheric pressure on the top of the valve Q forces it to its seat and checks the overflow. The jet of water then flows steadily through the delivery tube Y into the boiler through the pipe Z, as shown by the arrows. The injector is started by opening the globe valves in the steam and feed-water pipes. If the water supply is too great, water will

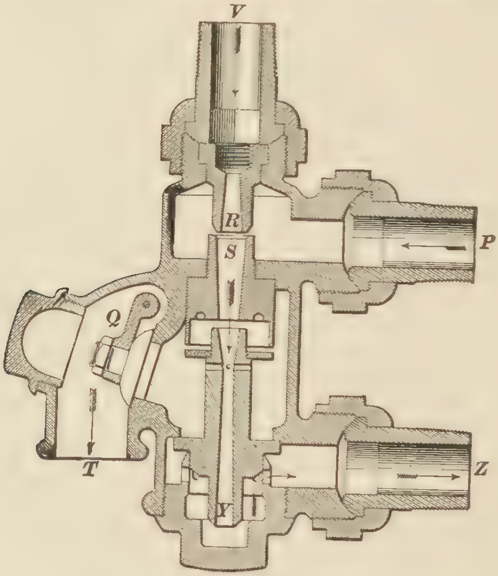


FIG. 591.

appear at the overflow, and the feed-valve must be closed a little or the valve in the steam pipe opened wider.

The capacity of an injector depends mainly upon the diameter of the delivery tube. The following table gives the capacities of the various sizes of the injector just described, for pressures from 45 to 75 pounds, and lifts from 1 to 6 feet:

TABLE 42.

Size.	Pipe Connections.			Capacity per Hour.		Horsepower of Boiler.
	Steam.	Suction.	Delivery.	Maximum.	Minimum.	
O O	$\frac{3}{8}$ in.	$\frac{3}{8}$ in.	$\frac{3}{8}$ in.	80 gal.	55 gal.	4 to 8
A	$\frac{1}{2}$ "	$\frac{1}{2}$ "	$\frac{1}{2}$ "	120 "	70 "	8 " 15
A A	$\frac{1}{2}$ "	$\frac{1}{2}$ "	$\frac{1}{2}$ "	165 "	90 "	15 " 20
B	$\frac{3}{4}$ "	$\frac{3}{4}$ "	$\frac{3}{4}$ "	250 "	135 "	20 " 30
B B	$\frac{3}{4}$ "	$\frac{3}{4}$ "	$\frac{3}{4}$ "	340 "	165 "	30 " 40
C	1 "	1 "	1 "	475 "	300 "	40 " 60
C C	1 "	1 "	1 "	575 "	350 "	60 " 72
D	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	750 "	400 "	72 " 93
D D	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	$1\frac{1}{4}$ "	920 "	500 "	93 " 120
E	$1\frac{1}{2}$ "	$1\frac{1}{2}$ "	$1\frac{1}{2}$ "	1300 "	700 "	120 " 160
E E	$1\frac{1}{2}$ "	$1\frac{1}{2}$ "	$1\frac{1}{2}$ "	1740 "	900 "	160 " 220
F	2 "	2 "	2 "	2270 "	1100 "	220 " 290
F F	2 "	2 "	2 "	2800 "	1400 "	290 " 350

It is considered good practice to provide the boiler with an injector considerably larger than actually needed, for, by throttling the feed, the injector will deliver much less than its maximum capacity, but will heat the water nearly to boiling point. The injector thus acts as a feed-water heater, and effects an economy of fuel.

An injector will not work efficiently when the feed is at high temperature. For example, the Penberthy injector will not work without overflowing when the feed temperature is above about 120°, or when the temperature of the delivery is above about 200°.

Injectors are used almost exclusively on locomotives, and to a large extent on stationary boilers, especially portable boilers and those of small dimensions.

The injector is an economical instrument, for, though but a small per cent. of the energy of the steam is used in actually forcing the water into the boiler, the remainder is not wasted, but is utilized in heating the feed.

1899. The **pumps** used for boiler feeding are usually direct-acting steam pumps, though in some cases power pumps are preferred. The former are independent, while the latter class are driven by belting from a line of shafting, and, hence, can not be used except when the engine is running. The power pump is more economical than the steam pump, but the independence and convenience of the latter have brought it into general use.

The direct-acting steam pump was described in Art. **1085.**

1900. The size of pump to feed a given boiler plant is easily determined by calculating the steam required per hour. This is done approximately by multiplying the rated horsepower of the boilers by 30, and adding, perhaps, 15 or 20 per cent. as a margin of safety. The pump should be large enough to feed the boiler under ordinary conditions by running continuously at slow speed, say 30 or 40 strokes per minute. Having given the feed-water consumption and

the number of strokes per minute, the size of the water cylinder is quickly found.

For example, suppose it is required to find the size of a pump to deliver 4,800 pounds of water per hour, the number of strokes per minute, under ordinary conditions, being 30 per minute.

There are $62\frac{1}{2}$ pounds of water in a cubic foot; hence, the water feed per *minute* in cubic feet is

$$\frac{4,800}{62\frac{1}{2} \times 60} = 1.28 \text{ cu. ft.}$$

The water displaced per stroke must, therefore, be

$$\frac{1.28}{30} \text{ cu. ft.} = \frac{1.28}{30} \times 1,728 = 73.73 \text{ cu. in.}$$

This would give a length of stroke of 6 inches and a cylinder diameter of 4 inches, nearly.

Other appliances, such as feed-water heaters, purifiers, economizers, etc., have already been described

A SERIES

OF

QUESTIONS AND EXAMPLES

Relating to the Subjects Treated of in Vol. II.

It will be noticed that, although the various questions are numbered in sequence from **614** to **958**, inclusive, these questions are divided into five different sections, corresponding to the five sections of the preceding pages of this volume. Under the heading of each section is given, in parentheses, the numbers of those articles which should be carefully studied before attempting to answer any question or to solve any example occurring in the section.

STEAM AND STEAM ENGINES.

(ARTS. 1189–1330.)

(614) (*a*) What is saturated steam; superheated steam? (*b*) In what way does saturated steam differ from a perfect gas?

(615) Explain fully the process of changing a quantity of water at 32° F. to steam.

(616) (*a*) Define the following: Total heat; heat of the liquid; latent heat of vaporization. (*b*) How may the "heat of the liquid" be found approximately?

(617) Find by formula **90**, the pressure corresponding to a temperature of 290° F. Ans. 57.92 lb. per sq. in.

(618) Find the total heat of vaporization of 1 pound of steam at a temperature of 318° F.

(619) Explain why saturated steam condenses when expanding in a non-conducting cylinder.

(620) If the pressure of the steam in a boiler is 81 pounds per sq. in. (absolute), what is the volume in cubic feet of 6 pounds of the steam? Ans. 31.7076 cu. ft.

(621) If a pound of steam occupies a volume of 7 cu. ft., what is its absolute pressure? Ans. 60.086 lb. per sq. in.

(622) Answer the following questions by referring to the steam table: (*a*) What is the gauge pressure of steam at 320° F.? (*b*) What is the temperature of saturated steam at a gauge pressure of 110 pounds per sq. in.? (*c*) What is the "heat of the liquid" and "total heat" of a pound of steam at 70 pounds per sq. in. (gauge)? (*d*) What is the

latent heat of vaporization of 3 pounds of steam at a pressure of 67 pounds, absolute?

Ans. (a) 75.18 lb. per sq. in.
(b) 343,951°.
(c) 286.003 B. T. U.; 1,178.266 B. T. U.
(d) 2,711.193 B. T. U.

(623) (a) What is the isothermal line of saturated steam? (b) What curve represents most closely the expansion of saturated steam?

(624) Find from the steam table the weight of 38 cu. ft. of steam at an absolute pressure of 43 pounds per sq. in.

Ans. 3.95466 lb.

(625) Show by a diagram why the expansive use of steam is economical.

(626) How much steam at 60 pounds gauge pressure must be mixed with 300 pounds of water at 55° F. in order that the temperature of the mixture may be 140° F.?

Ans. 23.885 lb.

(627) Find from the table the volume of $4\frac{1}{3}$ pounds of steam at a temperature of 256° F. Ans. 53.226 cu. ft.

(628) How many B. T. U. are required to raise 6 pounds of superheated steam from 310° to 342° F.?

Ans. 92.16 B. T. U.

(629) Calculate by formula **90**, the pressures corresponding to the following temperatures, and compare the results with those given in the steam table: (a) 254° F.; (b) 377° F.

Ans. (a) 32.094 lb. per sq. in.
(b) 190.54 lb. per sq. in.
From the tables:
(a) 32 lb. per sq. in.
(b) 189.212 lb. per sq. in.

(630) What is the action of steam in a cylinder of conducting material like cast iron? (b) What is the use of the steam jackets?

(631) Find by formula **91**, the total heat of vaporization of a pound of steam at the following pressures (absolute), and compare the results obtained with the steam table: (*a*) 210 lb. per sq. in.; (*b*) 88 lb. per sq. in.; (*c*) 37 lb. per sq. in.

Ans. { (*a*) 1,199.596 B. T. U.
(*b*) 1,179.086 B. T. U.
(*c*) 1,161.999 B. T. U.

(632) Show fully why the efficiency of the steam engine may be increased by raising the pressure of the steam used.

(633) Draw a "curve of constant steam weight" by laying off to some scale the volumes taken from column 6 of the table along the horizontal line $O X$ and the corresponding pressures as ordinates.

(634) (*a*) Define the following: Head end and crank end of cylinder; forward stroke; return stroke; reciprocating parts. (*b*) When is an engine said to "*run over*"? When does it "*run under*"? (*c*) What is the stroke, and to what is it equal?

(635) (*a*) Define the following: Clearance; real cut-off; apparent cut-off; ratio of expansion. (*b*) The clearance is 7% and the apparent cut-off $\frac{3}{8}$; find the real cut-off and the ratio of expansion.

Ans. (*b*) { Real cut-off, .416.
Ratio of expansion, 2.4.

(636) A non-condensing engine has a clearance of 6%, and cuts off at $\frac{3}{8}$ stroke. If it uses steam at a boiler pressure of 60 pounds per sq. in. (gauge), what is the probable M. E. P.?

Ans. 47.1 lb. per sq. in.

(637) (*a*) What is the I. H. P. of the engine of question 636, assuming the piston to be 11″ in diameter, the length of stroke 18″, and the number of revolutions 210? (*b*) If the mechanical efficiency of the engine is 83%, what is the net H. P.? (*c*) The friction H. P.?

Ans. { (*a*) 85.45 H. P.
(*b*) 70.92 H. P.
(*c*) 14.53 H. P.

(638) (*a*) Define lap; lead; angular advance. (*b*) How does the outside lap influence the "point of cut-off"? (*c*) What is the duty of the inside lap?

(639) What is an indicator diagram? (*b*) What three principal uses are made of them?

(640) (*a*) The area of an indicator diagram is 3.47 sq. in. One inch of the length of the diagram represents 6 inches of the length of stroke, the scale of the spring is 40, and the diameter of the engine piston is 16 in. What is the work of the engine per stroke? (*b*) What is the horsepower of the engine if it makes 120 R. P. M.?

Ans. { (*a*) 13,953.73 ft.-lb.
(*b*) 101.48 H. P.

(641) What are the faults of the diagram shown in Fig. 5?

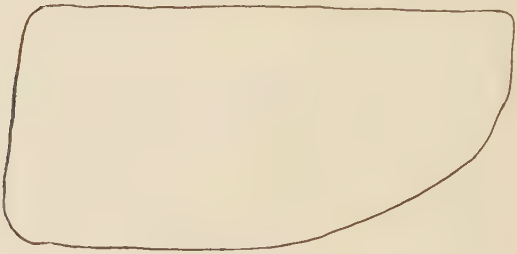


FIG. 5.

How might these faults be partially remedied?

(642) To drive the machinery of a certain shop requires 120 *actual* H. P. Find the probable dimensions of a Corliss condensing engine to do the work, assuming a boiler pressure of 70 pounds (gauge), and a ratio of expansion of 4. Assume a proper piston speed and mechanical efficiency.

(643) (*a*) What is the thermal efficiency of a steam engine using steam at 60 pounds gauge pressure, and exhausting at 2 pounds above the *atmosphere?* (*b*) If the pressure in the above case were raised to 90 pounds (gauge), and a condenser added, giving a back pressure of 3 pounds above *vacuum*, what would be the efficiency?

Ans. { (*a*) 11.55%.
(*b*) 23.93%.

(644) (*a*) The piston speed of an engine is 540 feet, the number of revolutions 150; what is the length of stroke?

(*b*) If the length of stroke is $2\frac{1}{2}$ ft. and the piston speed 900 ft. per minute, what is the number of revolutions?

Ans. $\begin{cases} (a)\ 21.6''. \\ (b)\ 180 \text{ R. P. M.} \end{cases}$

(645) Find the water consumption per I. H. P. per hour from the diagram shown in Fig. 6. Scale of spring, 40.

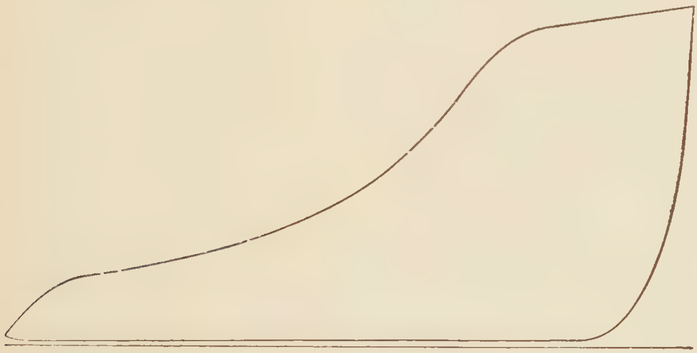


FIG. 6.

SUGGESTION.—Find the M. E. P. by dividing the diagram and measuring the ordinates, and then use formula **101**.

Ans. 22.76 lb. per I. H. P. per hour.

(646) (*a*) Explain fully the thermal advantages of the compound engine; (*b*) the mechanical advantage.

(647) The stroke of a tandem-compound engine is 30 inches. The diameter of the low-pressure cylinder is 32 inches, and of the high-pressure cylinder 19 inches. The M. E. P. of the high-pressure cylinder is 52 pounds, and of the low-pressure cylinder 18 pounds; the number of revolutions, 120. (*a*) Find the I. H. P. of the engine. (*b*) What is the ratio of the work done in each cylinder?

Ans. $\begin{cases} (a)\ 531.29 \text{ H. P.} \\ (b)\ 1.0184. \end{cases}$

(648) (*a*) What defects in the distribution of steam may be shown by the indicator diagram? (*b*) What may be done to remedy the following faults:

1. Admission, release, and compression too early?

2. Admission, release, and compression too late ?
3. Cut-off too early ?
4. Cut-off too late ?

(649) (*a*) Locate the points of cut-off, release, and compression on the diagrams of Fig. 7. (*b*) Assuming them

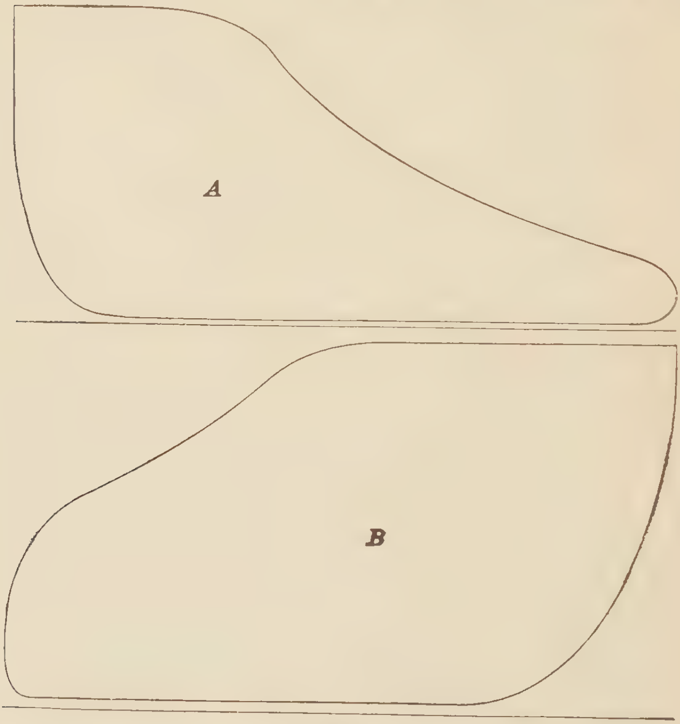


Fig. 7.

to have been taken with a 30 spring, find their average M. E. P.

(650) What should be the condensing surface of a surface condenser for an engine developing 675 I. H. P., and using $26\frac{1}{2}$ pounds of steam per I. H. P. per hour ?

Ans. 1,688.58 sq. ft.

(651) Given, a $30'' \times 48''$ engine making 80 revolutions per minute. On measuring a crank-effort diagram, it is found

that the fly-wheel is to store up at each stroke 88 foot-pounds of energy per sq. in. of piston area. Assuming a coefficient of unsteadiness of $\frac{1}{50}$, what must be the weight of the fly-wheel rim if the fly-wheel is 20 ft. in diameter?

Ans. 14,251.92 lb.

(652) A compound engine, developing 1,200 I. H. P., has a stroke of 42″, and a ratio of expansion of $5\frac{1}{2}$. The engine makes 70 R. P. M.; the total M. E. P. reduced to the low-pressure cylinder is 42 pounds per sq. in. (*a*) Find the diameter of the low-pressure cylinder. (*b*) Find the diameter of the high-pressure cylinder by formula **103**, and (*c*) by formula **104**.

Ans. $\begin{cases} (a)\ 49\frac{1}{2} \text{ in.} \\ (b)\ 34\frac{7}{8} \text{ in.} \\ (c)\ 32\frac{3}{8} \text{ in.} \end{cases}$

(653) (*a*) What will be the real cut-off in the high-pressure cylinder of problem 652, if its diameter is found by formula **103**? (*b*) If found by formula **104**?

Ans. $\begin{cases} (a)\ .368 \\ (b)\ .426 \end{cases}$

(654) For what is the Stephenson link used? Describe its action.

(655) The net pressure on the piston of a steam engine is shown by the diagram, Fig. 8. Draw the crank-effort diagram both to a semicircular base and to a straight base. Assume that the length of the connecting-rod is double the stroke.

FIG. 8.

(656) The steam enters a jet condenser at a pressure of $4\frac{1}{2}$ pounds above vacuum. The condensing water enters at a temperature of 55° and leaves, after mixing with the exhaust steam, at a temperature of 130°. What weight of condensing water must be used per pound of steam?

Ans. 13.76 lb.

(657) (*a*) Describe the action of the pendulum governor. (*b*) What is the distinction between the duty of a governor and that of a fly-wheel ?

(658) A steam engine is to develop 42 H. P. The M. E. P. available is 36.3 pounds per square inch; the engine is to run at 155 revolutions per minute. (*a*) Find the stroke, and (*b*) the diameter of piston, assuming the former to be $1\frac{1}{3}$ times the latter. (*c*) What is the piston speed of the engine ?

Ans. { (*a*) 15 in., nearly. (*b*) $11\frac{1}{4}$ in., nearly. (*c*) 387.5 ft. per min.

(659) (*a*) The diameter of an engine piston is 22″; the pressure at beginning of stroke is 72 pounds per square inch. What is the pressure on the crank-shaft ? (*b*) The crank now makes an angle of 60° with the axis of the cylinder, and the steam pressure has fallen to 65 pounds. What is the total pressure on the crank-shaft ? (*c*) What is the force tangent to the crank-pin circle ?

Ans. { (*a*) 27,369.62 lb. (*b*) 12,354.34 lb. (*c*) 21,398.25 lb.

(660) The M. E. P. of an engine obtained from the card is 34.6 pounds per square inch. What should be the mean ordinate of the crank-effort diagram expressed in the same units ? Ans. 22.03 lb.

(661) A locomotive with 19″ × 24″ cylinders and 80″ drivers uses steam at 150 gauge pressure, and is designed to run at a speed of 60 miles per hour. Under the ordinary load the M. E. P. is $52\frac{1}{2}$ lb. What I. H. P. does the locomotive develop ? Ans. 909.72 I. H. P.

(662) The travel of a valve is 5″; the lap is $\frac{3}{4}$″. (*a*) What must be the angular advance of the eccentric ? (*b*) What angle will the eccentric make with the axis of the cylinder when cut-off occurs ?

Ans. { (*a*) 17° 28′. (*b*) 72° 32′.

(663) The actual horsepower of an engine was experimentally found to be 12.325. The I. H. P. from the diagram was 15.36. (*a*) Find the efficiency of the engine.

(*b*) The engine had a $9'' \times 12''$ cylinder, and ran at an average speed of 240 rev. What was the M. E. P.?

Ans. $\begin{cases} (a) & 80.24\%. \\ (b) & 16.6 \text{ lb.} \end{cases}$

(664) Describe the steam-engine mechanism. What office does it fulfil?

(665) Draw the theoretical diagram of a simple steam engine from the following data: Boiler pressure, 70 pounds per sq. in. (gauge); ratio of expansion, 3; back pressure, 2 lb. above atmosphere; clearance, 7%. The steam is compressed to 40 lb. gauge pressure. Assume your own scale of pressures and volumes.

(666) What is a balanced valve, and why are they used?

(667) What is the object of computing the steam consumption from both the point of cut-off and a point near release?

(668) Calculate the size of a simple engine to give 240 *actual* H. P., choosing your own type of engine, a suitable boiler pressure, ratio of expansion, and mechanical efficiency.

(669) (*a*) What should be the scale of the indicator spring used if the boiler pressure is 54 pounds? (*b*) If the boiler pressure is 115 pounds? (*c*) The height of the steam line of a diagram above the atmospheric line is 1.834″. What is the gauge pressure of the steam at admission if the diagram was taken with a 40 spring?

(670) During an engine test 906 pounds of exhaust steam passed through the surface condenser, and 13,580 pounds of water were required for condensation. The average pressure of the exhaust steam was 7 pounds above vacuum. The condensing water entered at an average temperature of 52° F., and left the condenser at an average temperature of 120° F. What was the average temperature of the condensed steam on leaving the condenser? Ans. 148.66°.

(671) The diameters of the high-pressure, intermediate, and low-pressure cylinders of a triple-expansion engine are, respectively, 27″, 42″, and 66″. The M. E. P.'s of the steam

in the three cylinders are, respectively, 72 pounds, 40 pounds, and 16.5 pounds. The stroke of the engine is 4 feet, and the number of revolutions 70. (*a*) Find the I. H. P. of the engine. (*b*) Find the percentage of the work done in each cylinder.

Ans. (*a*) 2,598 I. H. P.
(*b*) 26.9% in high-pressure cylinder.
36.2% in intermediate cylinder.
36.9% in low-pressure cylinder.

(672) What are the distinctive features of the Corliss valve gear? What advantages has it over the plain slide valve?

(673) A Corliss condensing engine cuts off at $\frac{1}{4}$ stroke, and has a clearance of $2\frac{1}{2}$%. The boiler pressure is 84 pounds. (*a*) Find the probable M. E. P. (*b*) Assuming a piston speed of 500 feet per minute, what must be the diameter of the piston in order that the engine may develop 120 H. P.?

Ans. (*a*) 52.455 lb. per sq. in.
(*b*) $13\frac{7}{8}$ in., nearly.

(674) The gauge pressure of the steam in a boiler is 93 pounds. Find its temperature by formula **90.**

Ans. 332.609°.

(675) Draw a skeleton diagram showing the position of the crank and eccentric when the engine runs under, the angle of advance being 37°. The valve and cylinder may be omitted.

(676) Find, by means of formula **93,** the I. H. P. of an 18″ × 24″ engine, whose mean effective pressure is 62.4 pounds per square inch, and which makes 175 revolutions per minute.

Ans. 336.825 I. H. P.

(677) In Fig. 9 are shown the indicator diagrams taken from the high and low-pressure cylinders of a tandem compound non-condensing engine. Diameter of high-pressure cylinder, 13″; of low-pressure cylinder, 20″; stroke, 15″; revolutions per minute, 230. Find the horsepower.

Ans. 177.1 H. P.

(678) Take the clearance of both cylinders of the engine in the last example as 10% of the volume of the

high-pressure cylinder, and combine the diagrams into one, as in Fig. 297.

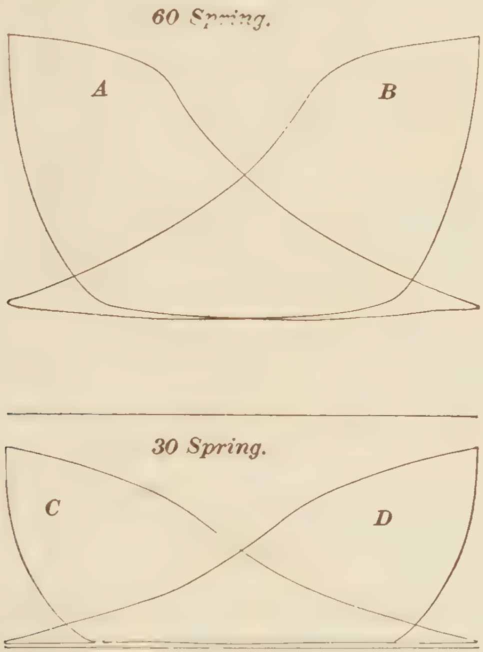


FIG. 9.

(679) Find the water consumption per I. H. P. per hour of a 24″ × 30″ engine making 150 revolutions per minute, and giving a diagram like *A*, Fig. 7. Scale of spring, 30.
Ans. 26.21 lb. per I. H. P. per hour.

(680) (*a*) What is meant by high rotative speed? (*b*) What is the difference between a high rotative speed and a high piston speed? (*c*) Does an engine with a high rotative speed necessarily have a high piston speed?

(681) The pressure on the steam gauge of a boiler is 123 lb.; find the temperature by formula **90.** Ans. 351.49°.

(682) A 300 horsepower engine has a cylinder 22″ in diameter, and a stroke of 18″; that is, it is a 22″ × 18″

engine. When making 200 revolutions per minute, what must be the mean effective pressure?

Ans. 43.406 lb. per sq. in.

(683) In a manner similar to question 675, find the position of the eccentric for an angle of advance of 20°, when the engine has a rocker arm to change the direction of motion of the valve.

(684) In Fig. 10 are shown the diagrams from both ends of the cylinder of a 13″ × 12″ engine running at 300 revolutions per minute. Find the net mean pressure upon the

FIG. 10.

pistons, as was done in Art. **1312**, by combining the expansion line of card A with the back-pressure line of card B. Scale of spring, 60. Ans. 36.96 lb. per sq. in.

(685) Find the indicated horsepower of the engine of example 684. Ans. 91.66 I. H. P.

(686) Find the water consumption per I. H. P. per hour from the diagrams of Fig. 10, example 684.

Ans. 21.49 lb. per I. H. P. per hour.

(687) Taking the ratio of the length of the connecting-rod to that of the crank as 5 : 1, draw the diagrams of tangential pressures on the crank-pin, both for a circular base and a straight base, of the engine of example 684.

(688) Calculate the weight of the fly-wheel rim of the engine in example 684, taking the coefficient of unsteadiness E as $\frac{1}{50}$, and the ratio n as 5. Ans. 623 lb.

STRENGTH OF MATERIALS.

(ARTS. 1331–1421.)

(689) (*a*) What are the advantages and disadvantages of cast iron as a material for construction purposes? (*b*) Of steel? (*c*) Of wrought iron? (*d*) For what purposes are each of the above largely used?

(690) Define the following: (*a*) Stress; (*b*) strain; (*c*) elasticity; (*d*) coefficient of elasticity; (*e*) ultimate strength.

(691) In how many ways may stress be applied to a body? Give an example of each.

(692) How much will a wrought-iron rod 2″ in diameter and 10 feet long be elongated by a pull of 40 tons?

Ans. .122 in.

(693) A steel rod $7\frac{1}{2}''$ long and $\frac{1}{2}''$ in diameter is elongated .009″ by a pull of 7,000 pounds; what is the coefficient of elasticity?

Ans. 29,708,853 in.-lb.

(694) What force is required to produce an elongation of .006″ in a cast-iron bar $1\frac{1}{2}'' \times 2''$ and 9 feet long?

Ans. 2,500 lb.

(695) A wooden rod 3″ in diameter is elongated .05″ by a force of 2,000 pounds; what was its original length?

Ans. 265.07 in.

(696) What is the minimum diameter of a wrought-iron bolt which is to withstand a steady pull of 6 tons with safety?

Ans. 1.054 in.

(697) How long must a cast-iron bar be, if supported vertically at its upper end, to break under its own weight?

Ans. 6,400 feet.

(698) The diameter of a wrought-iron bolt is $\frac{3}{4}''$; what should be the thickness of the bolt head in order that the bolt may be equally strong in tension and in shear; in other words, what should be the thickness of the head in order that the force required to pull the bolt apart shall be just equal to the force required to strip the head from the bolt?

Ans. .206 in.

(699) What safe load may be borne by a brick pier, the cross-section of which is $2\frac{1}{2} \times 3\frac{1}{2}$ feet? Ans. 105 tons.

(700) In Fig. 11, the force P is $1\frac{3}{4}$ tons and acts at an angle of 30° with the horizontal. The width b of the timber tie-rod is 4 inches. Find the safe length a necessary to prevent shearing on the surface a b.

Ans. 10.1 in.

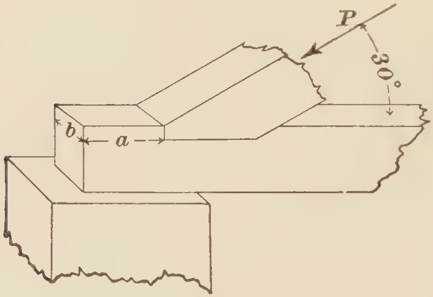


FIG. 11.

(701) Show that a cylindrical boiler is twice as liable to rupture along a longitudinal seam as along a transverse seam.

(702) Assuming that the strength of the boiler plate is reduced 40% owing to the area lost by the rivet holes, what should be the thickness of the plate of a wrought-iron boiler 4 feet in diameter which sustains a steam pressure of 120 lb. per sq. in.? Assume factor of safety for steady stress.

Ans. .35 in.

(703) What should be the thickness of a 6″ cast-iron water pipe to carry a steady pressure of 200 lb. per sq. in.?

Ans. .18 in.

(704) What should be the thickness of a wrought-iron boiler tube 3″ in diameter, 12 ft. long, and exposed to an external steam pressure of 130 lb. per sq. in.? Use a factor of safety of 10. Ans. .272 in.

(705) A cast-iron cannon of 4″ bore is subjected to an internal pressure of 2,000 lb. per sq. in. on being fired; what should be the thickness of the metal in order that it shall not be subjected to a stress of more than 2,800 lb. per sq. in.?

Ans. 5 in.

(706) A structure of the nature of a simple beam is 100 feet long between the supports, and carries three equal loads of 1,200 lb. each, at distances from the left support of 40, 60, and 80 ft. (*a*) Find the maximum bending moment; (*b*) the shear at a distance of 30 ft. from the left support; (*c*) the maximum shear.

Ans. { (*a*) 748,800 in.-lb. (*b*) 1,440 lb. (*c*) – 2,160 lb.

(707) Find graphically the magnitude, direction, and position of the resultant of the forces shown in Fig. 12.

NOTE.—In Figs. 12 and 14, the line mn is drawn simply to locate the forces. Thus, in Fig. 12, F_1 makes an angle of 47½° with mn; F_2 makes an angle of 72½° with mn, and so on. The distance between the intersection of F_1 and F_2 with mn is 1 inch; the distance between the intersections of F_3 and F_2 is 1¼ inches, etc.

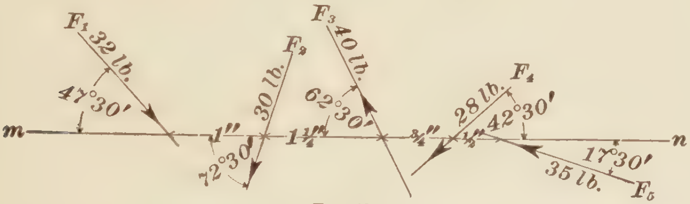


FIG. 12.

(708) A beam is loaded as in Fig. 13. Find the reaction of the supports, and draw the shear diagram.

(709) (*a*) Where is the maximum bending moment of the above beam? (*b*) What is it in inch-tons? (*c*) In inch-pounds?

Ans. { (*b*) 288 in.-tons. (*c*) 576,000 in.-lb.

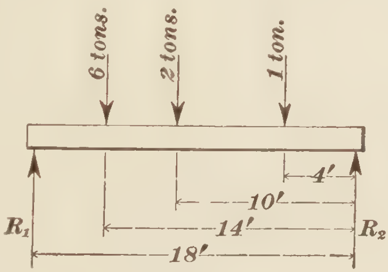


FIG. 13.

(710) Define the following: Simple beam; cantilever; restrained beam; continuous beam.

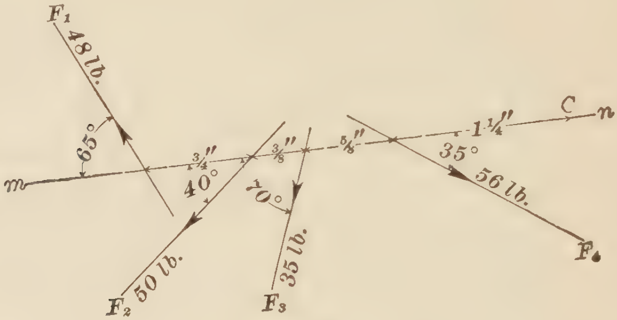


FIG. 14.

(711) Find the resultant moment of the forces of Fig. 14 about the point C.

(712) A rectangular white-pine beam 22 feet long carries a load of 4 tons, 8 feet from one end. Assuming the depth of the beam to be double the width, what should be its dimensions to safely support the load ?

Ans. Depth = $17\frac{1}{3}$ in.; breadth = $8\frac{2}{3}$ in.

(713) The beam of Fig. 15 is loaded as shown and has, in addition, a uniform load of 40 pounds per foot. (a) Find the maximum shear; (b) the maximum bending moment; (c) the reactions of the supports.

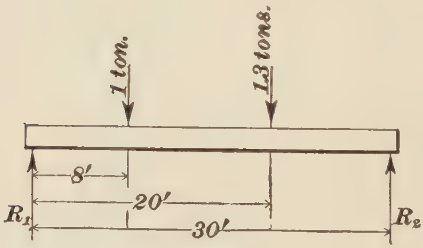


FIG. 15.

Ans. { (a) 2,930 lb.; (b) 320,000 in.-lb.; (c) 2,930 and 2,870 lb. }

(714) The moment of inertia of a wrought-iron I beam 12″ deep is 280; what uniform load will it sustain ? The stress is assumed to be steady. The beam is 20 ft. long and rests upon two supports.

Ans. 875 lb. per ft.

(715) A hollow circular cast-iron beam 8 feet long rests upon two supports. The inside diameter is 5 in. and the outside diameter $6\frac{1}{2}$ in.; what is the maximum safe load that may be concentrated at its center? Ans. 4,624 lb.

(716) A hemlock floor beam is $2'' \times 10''$ and 16 feet long; (*a*) what distributed load will it carry? (*b*) What distributed load would it carry if laid with the 10″ side horizontal?

Ans. $\begin{cases} (a) & 78.12 \text{ lb. per ft.} \\ (b) & 15.6 \quad \text{"} \quad \text{"} \end{cases}$

(717) Find the deflections of the beams of (*a*) example 714, (*b*) example 715, and (*c*) example 716 (*a*).

Ans. $\begin{cases} (a) & .45 \text{ in.} \\ (b) & .1 \text{ in., nearly.} \\ (c) & .461 \text{ in.} \end{cases}$

(718) The piston of a steam engine is 14 in. in diameter; the steam pressure is 80 lb. per sq. in. Assuming that the total pressure on the piston comes on the crank-pin at the dead points, and considering the crank-pin as a cantilever uniformly loaded, what should be its diameter if 4 inches long and made of wrought iron? Take a factor of safety of 10. Ans. 3.82 in.

(719) What load can be safely sustained by a cast-iron column 14 feet long and 6″ in diameter, with flat ends?

Ans. 120,865 lb.

(720) What should be the dimensions of a piece of timber 30 ft. long, with flat ends, to support a load of 7 tons, the cross-section being in the form of a square?

Ans. $9\frac{5}{8}$ in. square.

(721) A cylindrical steel connecting-rod $7\frac{1}{2}$ ft. long is subjected to a maximum stress of 21,000 pounds. Assuming it to be a column with both ends hinged, and subjected to shocks, determine its diameter. Ans. $2\frac{5}{8}$ in.

(722) A wrought-iron piston rod has a diameter of 2″ and a length of 4 feet. Assuming it to be a column with one end flat and the other rounded, what is the allowable diameter of the piston if the steam pressure is 60 pounds per sq. in.? The rod is subjected to shocks. Ans. $15\frac{3}{4}$ in.

(723) A beam is $6'' \times 8''$ and 10 feet long, the $8''$ side being vertical. Another beam of the same material is $4'' \times 12''$ and 16 feet long, the $12''$ side being vertical. (*a*) What is the ratio between the maximum loads the two beams are capable of supporting? (*b*) What is the ratio between the deflections in the two cases, the manner of loading being the same?

Ans. $\begin{cases} (a) \ 1\frac{1}{15} : 1. \\ (b) \ .549 : 1. \end{cases}$

(724) (*a*) What should be the diameter of a round wrought-iron shaft to transmit 40 H. P. at 120 R. P. M.? (*b*) To transmit 80 H. P. at 100 R. P. M.?

Ans. $\begin{cases} (a) \ 3.739 \text{ in.} \\ (b) \ 4.65 \text{ in.} \end{cases}$

(725) What must be the diameter of a steel engine shaft to transmit 4,000 H. P. at 50 R. P. M.? Ans. 14.22 in.

(726) What H. P. can be transmitted by a steel shaft $4''$ square, making 80 R. P. M.? Ans. 71.775 H. P.

(727) What horsepower can be transmitted by a hollow wrought-iron shaft making 100 R. P. M., the outside diameter being $7\frac{1}{2}$ in., and the inside diameter 5 in.?

Ans. 717.7 H. P.

(728) (*a*) What weight may be safely sustained by a hemp rope $8''$ in circumference? (*b*) What should be the diameter of an iron wire rope of 7 strands to safely sustain a stress of 6,000 pounds? (*c*) What should be the circumference of a steel hoisting rope which is required to lift a load of $6\frac{2}{3}$ tons?

Ans. $\begin{cases} (a) \ 6{,}400 \text{ lb.} \\ (b) \ 1.054 \text{ in.} \\ (c) \ 3.65 \text{ in.} \end{cases}$

(729) (*a*) An open link chain is made of $\frac{7}{8}''$ iron; what safe load will it sustain? (*b*) What should be the diameter of the iron composing the links of a stud-link chain which is to sustain a load of 4 tons?

Ans. $\begin{cases} (a) \ 9{,}187.5 \text{ lb.} \\ (b) \ .667 \text{ in.} \end{cases}$

(730) A steel shaft $2''$ in diameter is supported by hangers 10 feet apart; if a pulley be placed midway between the hangers, what should be the maximum allowable belt pull on

the pulley in order that the shaft shall not deflect more than $\frac{1}{8}''$? Ans. 327.25 lb.

SUGGESTION.—Assume the shaft to be a restrained beam, loaded in the middle.

(731) An engine shaft rests on bearings 5 feet apart; midway between the bearings the shaft carries a fly-wheel weighing 3 tons. (*a*) Find the size of the shaft to withstand this load. (*b*) Find the size of the shaft on the assumption that the engine makes 80 R. P. M. and develops 75 H. P. The shaft is made of steel. Factor of safety, 10.

Ans. $\begin{cases} (a)\ 4\frac{1}{4} \text{ in., nearly.} \\ (b)\ 4\frac{5}{8} \text{ in., nearly.} \end{cases}$

(732) A white oak cantilever is loaded as shown in Fig. 16, and has in addition a uniform load of 80 pounds per foot. (*a*) Determine graphically the shear diagram and the maximum bending moment. (*b*) Find the necessary dimensions of the cross-section of the beam, assuming the depth to be $2\frac{1}{2}$ times the breadth, and the stress to be steady.

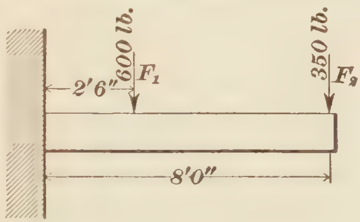


FIG. 16.

Ans. $\begin{cases} (a)\ 82{,}250 \text{ in.-lb.} \\ (b)\ b = 3.7 \text{ in.};\ d = 9\frac{1}{4} \text{ in.} \end{cases}$

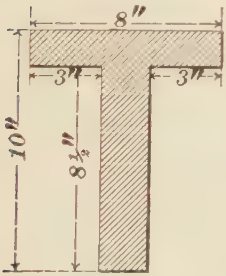


FIG. 17.

(733) A steel beam of the cross-section shown in Fig. 17 rests upon two supports 35 feet apart; (*a*) what concentrated load will it sustain at the center, the stress being considered as varying? (*b*) What total uniform load will it sustain, the stress being steady?

Ans. $\begin{cases} (a)\ 7{,}246 \text{ lb.} \\ (b)\ 20{,}290 \text{ lb.} \end{cases}$

(734) (*a*) The moment of inertia of a rectangle is 72; the area of the rectangle is 24 sq. in.; what is the value of r in formula **115**? (*b*) What are the dimensions of the rectangle? (*c*) The moment of inertia of a circle about an axis

through its center is $\frac{\pi d^4}{64}$; the area of the circle is $\frac{\pi d^2}{4}$; what is the radius of gyration of the circle?

Ans. $\begin{cases} (a) \ 1.732. \\ (b) \ b = 4 \text{ in.}; d = 6 \text{ in.} \\ (c) \ \frac{1}{4}d. \end{cases}$

(735) A cast-iron sphere 8″ in diameter is subjected to a steady internal pressure of 100 lb. per sq. in.; what should be its minimum thickness to safely withstand this pressure?

Ans. .06 in.

(736) A beam 28 feet long, uniformly loaded with 60 pounds per foot, rests upon two supports which are placed 5 feet from each end. Determine graphically the maximum shear and the maximum bending moment.

Ans. Bending moment = 20,160 in.-lb.

(737) Assuming the above beam to be made of white pine, find the dimensions of its cross-section, the stress being steady. The beam is to be rectangular.

(738) A wrought-iron boiler tube $2\frac{1}{2}$″ in diameter and 9 feet long has a diameter of .2″; what external pressure can the tube withstand, using a factor of safety of 10?

Ans. 106.43 lb. per sq. in.

(739) A crane chain is required to lift 5 tons; what should be the diameter of the iron composing the links of the chain, which is of the open link variety? Ans. .913 in.

(740) A gear-wheel 4 feet in diameter is keyed to a shaft. The force acting tangent to the pitch circle of the gear-wheel is 350 pounds; what should be the diameter of the shaft if made of steel? Ans. 2.84 in., say $2\frac{7}{8}$ in.

(741) The head of an engine cylinder, 12″ in diameter, is fastened on by 10 wrought-iron bolts. In order to make the joint steam tight, a safe stress of only 2,000 pounds per sq. in. is allowed. The steam pressure being 90 pounds per sq. in., what should be the minimum diameter of the bolts, that is, the diameter at the root of the thread? Ans. .8 in

(742) A beam 18 feet long has a load of 1 ton placed at each end. The supports are placed 5 feet from each end. (See Fig. 18.) (*a*) Draw the shear diagram and find the maximum bending moment. (*b*) If made of cast iron and circular in cross-section, what should be its diameter, the stress being steady?

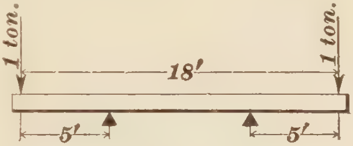


FIG. 18.

Ans. { (*a*) 120,000 in.-lb. (*b*) 5.784 in.

(743) A beam 2″ × 6″ in cross-section and 18 feet long deflects .3″; how much will a beam 3″ × 8″ and 12 feet long deflect under the same load? The long side is placed vertically in both cases. Ans. .025 in.

(744) A wrought-iron key, or cotter, is used to fasten a rod, as shown in Fig. 19. There is a pull of 20,000 pounds on the rod. Assuming the breadth of the key to be four times the thickness, find its dimensions to safely resist the stress. Use a factor of safety of 10.

Ans. { Breadth, 2.828″. Thickness, .707″.

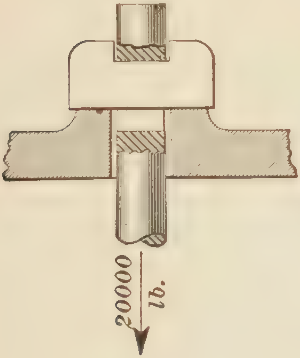


FIG. 19.

(745) A steel engine shaft 12″ in diameter, resting in bearings 54″ apart, carries a fly-wheel weighing 30 tons, midway between the bearings; what is the deflection of the shaft? Consider the shaft as a simple beam. Ans. .0064 in.

(746) (*a*) A 7-strand steel wire rope has a diameter of $1\frac{1}{4}$″; what load will it carry safely? (*b*) Find the circumference of a hemp rope to carry $1\frac{3}{4}$ tons.

Ans. { (*a*) 14,062.5 lb. (*b*) 5.92 in.

(747) What stress would be required to rupture a cast-iron cylinder which is 8″ in diameter and 6″ thick?

Ans. 12,000 lb.

(748) A beam which is 100 feet long carries four equal loads of 1,200 pounds each at distances of 20, 40, 60, and 80 feet from the left support. (*a*) Find the maximum bending moment, and (*b*) the shear at both supports.

Ans. $\begin{cases} (a) \text{ 864,000 in.-lb.} \\ (b) \text{ 2,400 lb.} \end{cases}$

APPLIED MECHANICS.

(ARTS. 1422–1588.)

(749) A lever is required to transfer motion from one line to another one parallel to it, and $4\frac{1}{2}''$ distant. The motion along one line is to be $3\frac{3}{4}''$, and along the other line $2\frac{3}{4}''$. Determine the length of the lever arms graphically, assuming the fulcrum to be between the ends of the lever.

(750) A motion of 6 inches along one line is to be transferred and increased to a motion of 8 inches along another line, making an angle of 45° with the first line. The direction of the motion is to be reversed. Make a drawing of a bell-crank lever, one-fourth size, which will accomplish this.

(751) Design a bell-crank lever like that shown in Fig. 347. Given, angle between center lines of motion, 10°; ratio of motions, 4 : 3; length of long arm, 16 inches. Draw one-fourth size.

(752) Design a lever indicator reducing motion which will give an exact reduction. Stroke of engine, 3 feet; length of card, $3\frac{1}{2}$ inches.

(753) A vibrating toothed pinion 12″ in diameter gears with a rack which has a corresponding reciprocating motion. How far must the rack move to cause the wheel to vibrate through an angle of $6\frac{1}{2}°$? Ans. .68 inch.

(754) Draw a skeleton diagram of the mechanism shown in Fig. 20; i. e., a diagram showing the positions of the center line of each link when in mid-position and in each extreme position. Crank *A* is the driver; the bell-crank *B* has equal arms of any convenient length; the long arm of the lever *C* makes an angle of 30° with the vertical when in

mid-position, and on its upper end carries a segment, gear ing with the pinion D. The pinion is to vibrate exactly

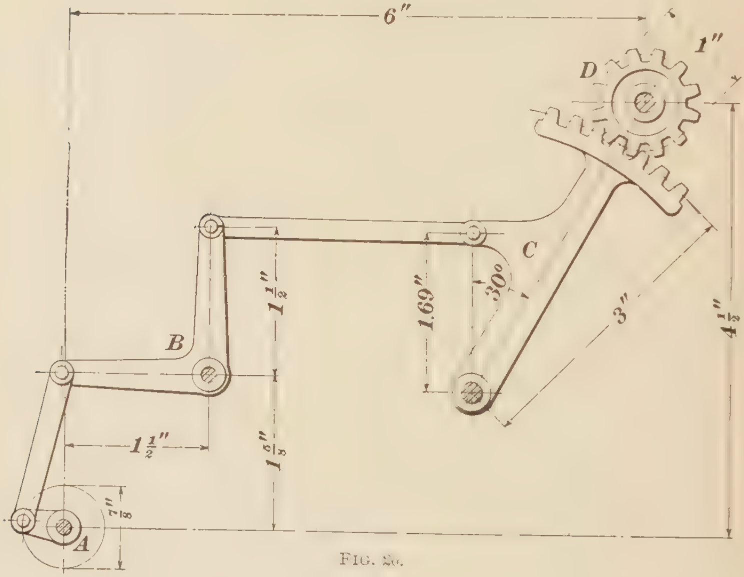


FIG. [illegible].

one-half a turn for each stroke of the lever C. All necessary dimensions are given in the figure.

(755) In the connecting-rod mechanism of Fig. 349, (*a*) does the cross-head move more rapidly during the first or during the second half of the return stroke? (*b*) What effect does increasing the length of the connecting-rod have upon this motion?

(756) Two cranks of the same length are keyed to the same shaft so as to be parallel, and revolve together, side by side; one crank drives a cross-head A by means of a connecting-rod, and the other crank drives a slotted cross-head B. If the crank turns *left-handed* and cross-head A is at the *left* of the crank-shaft, (*a*) which cross-head leads during the forward stroke? (*b*) Which, during the return stroke?

(757) In Fig. 21 is a bookbinder's press, consisting of two equal armed toggle-joints, drawn together by a right and

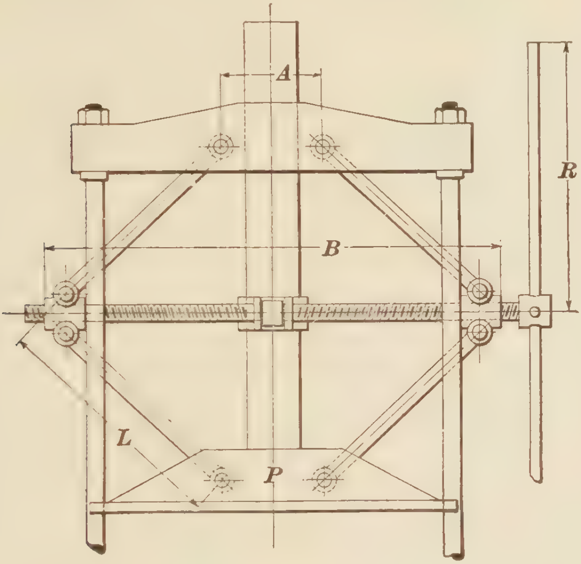


FIG. 21.

left hand screw. If $R = 10''$, $L = 18''$, $A = 12''$, and the screw has four threads per inch, what pressure would the platen P exert when a force of 75 lb. is exerted at the end of the handle, and the distance B is 16 inches? Neglect friction. Ans. 84,295.4 lb.

(758) Fig. 22 illustrates a mechanism in a certain machine. A pulley P drives shaft O, to which is attached crank disk D, the radius of the crank being $O\,R$. $C\,E$ is a vibrating lever with a pin E, at the end of which the connecting-rods $R\,E$ and $K\,E$ are attached, thus transmitting the motion to the cross-head H. With a belt pull of 2 lb., it is desired to find (a) the force exerted upon the pin K, and (b) the horizontal thrust exerted by the cross-head H, when $C\,E$ makes an angle of 30° with the horizontal.

As the example is to illustrate the principle only, the dimensions are, for convenience, made small.

Given, diameter of pulley, 3″; radius $O R = \frac{1}{2}''$; radius $C E = 1\frac{3}{4}''$; $K E = 5\frac{1}{2}''$. The length of $R E$ is such that the crank disk will cause point E to move exactly to E'.

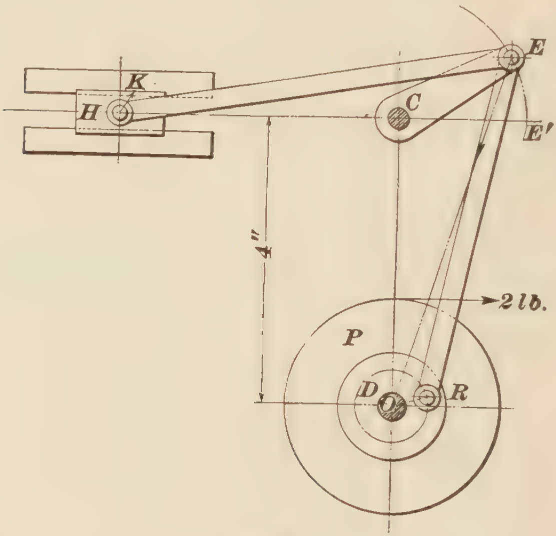


FIG. 22.

NOTE.—Draw $O E$ and consider $E O R$ as a crank and connecting-rod mechanism, with the cross-head at E, and moving in the direction of the arrow.

Ans. $\begin{cases} (a) & 12.8 \text{ lb.} \\ (b) & 12.6 \text{ lb.} \end{cases}$

(759) What adjustments to the stroke are necessary in a shaping or slotting machine where the tool has a reciprocating motion?

(760) (*a*) Show graphically how to construct a vibrating link motion, so that the periods of advance and return are to each other as 3 is to 2. Length of stroke, 24″; scale, 3″ = 1 ft. (*b*) What would be the effect of shortening the stroke of the tool slide?

(761) (*a*) Show how to construct a Whitworth quick-return motion, so that the period of return equals one-half of the period of advance. (*b*) Plot the motion for positions of the driving gear at every 20° of its motion. (*c*) What do

you consider to be the relative merits of the vibrating link and Whitworth quick-return motions?

(762) (*a*) A universal joint is used to couple two shafts at an angle of 28° with each other. If the driving shaft revolves 50 times per minute, what would be the greatest and least rates of motion of the driven shaft? (*b*) What would be the effect of using two universal joints, if the forks on the middle shaft which connects them *were at right angles with each other?*

Ans. (*a*) Greatest = 56.63 rev. Least = 44.15 rev.

(763) Given the stroke of a Watts straight-line motion = 2 ft.; distance from the center of the lower lever to the center line of motion of the guided point = 30″; perpendicular distance between the two levers when in mid-position = 3 ft.; perpendicular distance of the guided point from the lower end of the connecting link = 15″. Required the lengths of the two levers.

Ans. $\begin{cases} \text{Length of upper lever, 23.1″.} \\ \text{Length of lower lever, 31.2″.} \end{cases}$

(764) Find the outline of a cam to turn uniformly right handed and give motion to a point moving on a line passing through the axis of the cam. The point is to remain stationary during the first $\frac{1}{12}$ revolution of the cam; to move uniformly away from the center of the cam 2″ for the next $\frac{8}{12}$ revolution, and then to move to its first position with a harmonic motion during the remaining $\frac{3}{12}$ revolution. Take $1\frac{1}{2}$″ as the shortest distance between the point and the axis of the cam.

(765) Draw the outline for a cam to turn uniformly left-handed, and give a motion to a roller 1″ in diameter in a horizontal line, passing 1″ above the axis of the cam. The roller is to have a uniform motion of $2\frac{1}{2}$″ to the left during the first $\frac{1}{2}$ revolution of the cam; during the next $\frac{1}{4}$ revolution it is to remain at rest, and then is to move suddenly to its first position, where it is to remain during the last quarter-turn of the cam. The nearest position of the center of roller to the axis of the cam is to be 2″.

(766) A cylindrical cam 3″ in diameter is designed to move a roller uniformly to the left a distance of 1″ during the first half revolution; to move it uniformly to the right $\frac{1}{2}$″ during the next $\frac{1}{8}$ revolution, at which point it remains during the next quarter revolution. During the last $\frac{1}{8}$ of a revolution, it returns uniformly to its first position. The cam turns left-handed as looked at from the right. Draw the development of the groove, the diameter of the roller being $\frac{1}{2}$″.

(767) Lay out the outline for a positive-motion plate cam, like that in Fig. 369. Diameter of rollers, $\frac{3}{4}$″; distance between centers of rollers, 5″; required movement of rod, $1\frac{1}{4}$″.

(768) A pulley 4 ft. in diameter drives another pulley 52″ in diameter by means of a belt. If the speed of the driven shaft is 320 revolutions per minute, what is the speed of the driving shaft? Ans. $346\frac{2}{3}$ rev. per min.

(769) The driving pulley on a machine is 10″ in diameter, and should run at 500 revolutions per minute. What should be the diameter of the pulley on the countershaft, the speed of which is 210 revolutions per minute? Ans. $23\frac{4}{5}$″.

(770) A saw arbor is fitted with a pulley $3\frac{1}{2}$ inches in diameter. The countershaft has pulleys 20″ and 6″ in diameter. What size pulley should be placed on the main shaft, which runs at 189 revolutions per minute, to drive the saw at 1,800 revolutions per minute? Ans. 10 inches.

(771) A machine is to be belted through a countershaft so as to run at 900 revolutions per minute, the speed of the main driving shaft being 150 revolutions per minute. Find two ratios that could be used for each pair of pulleys.

(772) A shaft making 200 revolutions per minute has upon it a pulley 36″ in diameter, carrying a belt upon its periphery which transmits 5 horsepower. (*a*) What is the effective pull on the belt? (*b*) What diameter should the pulley have in order that the effective pull may be 50 lb.?

Ans. $\begin{cases} (a)\ 87.54 \text{ lb.} \\ (b)\ 63 \text{ inches} + . \end{cases}$

(773) Determine by the aid of Table 33 what you would consider to be a suitable width of single belt for driving a saw arbor under the following conditions: Greatest horsepower required, 6; diameter of smaller pulley, 4″; diameter of driving pulley, 30″; distance between the centers of the two pulleys, 5 ft.; number of revolutions of the saw arbor, 1,500.

(774) A 250-horsepower Corliss engine makes 90 revolutions per minute, and has a 14-ft. band wheel. How wide should the rim of the wheel be, the rim being 1″ wider than the belt. Calculate for a double belt. Ans. 41 inches.

(775) Two shafts connected by a 6-inch single belt running over 24-inch pulleys make 205 revolutions per minute. How much power should the belt transmit?

Ans. 8.6 H. P., nearly.

(776) If the belt were found to be inadequate for the work in the last example, which would be the more effectual remedy—to use a 6-inch double belt in place of the single belt, or to substitute 30″ pulleys for the 24″ pulleys, and still use a 6-inch single belt?

(777) Two continuous cones are required to give a range of speed between 70 and 400 revolutions per minute. Assuming the large diameters of the cones to be 18″, (*a*) what must be the small diameters? (*b*) What should be the speed of the driving shaft? Both cones are to be alike.

Ans. $\begin{cases} (a) & 7.53 \text{ in.} \\ (b) & 167\frac{1}{3} \text{ rev.} \end{cases}$

(778) In the above example, if the speed of the driving shaft were 225 revolutions per minute, (*a*) what should be the ratio of the large and small diameters, assuming the highest speed of the driven cone to be 400 revolutions per minute? (*b*) Thus designed, what would be the slowest speed at which the driven cone could run?

Ans. $\begin{cases} (a) & 1.78 : 1. \\ (b) & 126.56 \text{ rev.} \end{cases}$

(779) A pair of stepped cones, having 4 steps, each of 2-inch face, are required to be used with an open belt.

Given, largest diameter, 15″; smallest diameter, $4\frac{1}{2}$″; distance between centers, 20″; width of each step, 2″. Make an outline drawing of one of the cones, half size.

(780) Make a sketch showing how you would lace an 8-inch belt. Show the location of the holes by dimension figures.

(781) Two shafts lie in parallel planes, one above the other, but cross each other at an angle of 45°. Make a sketch showing how they might be connected by belt. Show two views, either a plan and elevation, or front and side elevation, and indicate by arrows the directions in which the shafts are to run.

(782) Two shafts lie in the same vertical plane, but make an angle of 30° with each other. Show how to connect them by belt so that they will revolve in opposite directions, and will run either backwards or forwards. Indicate the directions of rotation by arrows.

(783) A pulley A, upon a main shaft, drives a pulley B by means of a crossed belt. A spur gear C, on the shaft with B, drives a pinion D. A pulley E, on the shaft with D, drives a pulley F by an open belt.

Given, A, 30″ in diameter and making 60 revolutions per minute; B, 15″ diameter; C, 60 teeth; D, 15 teeth; E, 30″ diameter, and F, 10″ diameter. How many revolutions does pulley F make per minute, and in what direction does it turn relatively to A? Ans. 1,440 revolutions.

(784) A train of four gears is arranged as follows: Gear A drives gear B; gear B drives gear C; gear C drives D. A has 90 teeth and turns right-handed; B has 40 teeth; C has 80 teeth, and D, 90 teeth. How many turns does gear D make to every turn of A, and in what direction?

(785) Suppose the lead screw of a lathe to have a left-handed thread, the pitch being $\frac{1}{2}$″, and suppose the stud T (Fig. 398) to make $\frac{1}{2}$ as many turns as the spindle d_1 having 27 teeth, and f_1, 54 teeth. Find a set of change gears, the smallest gear to have 24 teeth, and the largest 96 teeth, to

cut threads from 4 to 16 per inch (including $11\frac{1}{2}$ per inch), making the number of gears as small as possible.

Ans.

Threads per Inch.	Gear on Stud.	Gear on Lead Screw.
4	96	24
5	96	30
6	96	36
7	96	42
8	96	48
9	96	54
10	48	30
11	48	33
$11\frac{1}{2}$	96	69
12	48	36
13	48	39
14	48	42
15	48	45
16	48	48

(786) A mangle rack and wheel are so proportioned that 5 turns of the wheel will produce one complete forward and back movement of the rack. If the gear turns uniformly, and has a pitch diameter of 10″, (*a*) how far will the rack travel one way with a *uniform* velocity? (*b*) What is the total travel of the rack one way? Ans. { (*a*) 5 ft. 2.832 in. (*b*) 6 ft. .832 in.

(787) In the clutch gearing, shown in Fig. 403, suppose the worm gear W to have 40 teeth; gear D_1, 80 teeth; gear F_1, 20 teeth, and the two bevel gears to be of the same size. If the worm is single-threaded, what is the ratio of the "quick return" through the spur and bevel gears to the slow motion through the worm and worm-wheel? Ans. 160 : 1.

(788) In Fig. 404, let the diameter of D_1 be $3\frac{1}{2}''$; of F_1, $26''$; of D_2, $3\frac{1}{2}''$. What is the ratio of the belt to cutting speed, supposing the driving pulleys to be 30″ in diameter? Ans. 63.6 : 1.

(789) In the differential train shown in Fig. 406, let wheel D have 100 teeth and remain stationary. (*a*) If wheel F has 99 teeth, and A revolves $+ 10$ times, how many

turns will F make, and in what direction? (*b*) If F should have 101 teeth, under the same conditions, how many turns would F make, and in what direction?

Ans. $\begin{cases} (a) & -.101 \text{ turn.} \\ (b) & +.099 \text{ turn.} \end{cases}$

(790) In the same train, let D have 50 teeth and F 40 teeth. (*a*) If D makes $+1$ turn and $A+4$ turns, how many turns does F make, and in what direction? (*b*) If D makes -1 turn and $A+5$ turns, how many turns does F make?

NOTE.—Consider the train locked as usual. Fix arm and turn D back to the starting point; then, give D the stated number of turns, with the arm still fixed.

Ans. $\begin{cases} (a) & +\frac{1}{4} \text{ turn.} \\ (b) & -2\frac{1}{2} \text{ turns.} \end{cases}$

(791) In Fig. 407, let F_1 have 20 teeth; D_1, 40 teeth; F, 20 teeth, and D, 200 teeth. If A makes $+6$ turns, how many turns does F_1 make? Ans. $+126$ turns.

(792) In the differential train shown in Fig. 23, pulleys A and B are connected by open belt. A and E are fast to

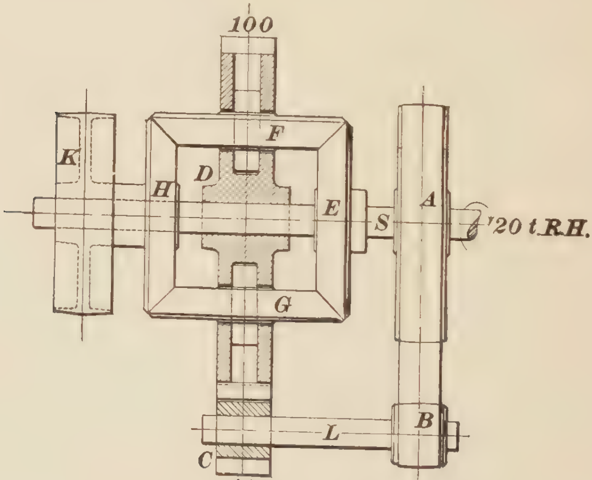


FIG. 23.

shaft S; D and H, and K are loose on S; H and K being one piece. Wheel D serves as an arm to carry gears F and G. B and C are keyed to shaft L.

Let H, F, E, and G each have 50 teeth; D, 100 teeth, and C, 10 teeth, and let the diameter of A be 12″ and of B 3″. If shaft S makes + 20 turns, as indicated by the arrow, how many turns will pulley K make, and in what direction?

NOTE.—Calculate as though E were stationary at first. Then, fix arm D, and turn E + 20 times.

Ans. − 36 revolutions.

(793) In designing gear teeth, what object should be accomplished, (*a*) with regard to the motion transmitted, and (*b*) with regard to the shape and contact of the teeth?

(794) (*a*) What are the relative advantages of circular and diametral pitch? (*b*) The diametral pitch of a gear is $4\frac{1}{2}$; what is the circular pitch? (*c*) The circular pitch of a gear is 1.1424″; what is the diametral pitch?

Ans. { (*b*) .698 inch. (*c*) $2\frac{3}{4}$.

(795) (*a*) Of what diameter is a 3-pitch gear having 30 teeth? (*b*) What should be the outside diameter of this gear? (*c*) What should be the working depth of the teeth? (*d*) The total depth?

Ans. { (*a*) 10 inches. (*b*) 10.667 inches. (*c*) .667 inch. (*d*) .709 inch.

(796) Define addendum; face; root; flank.

(797) (*a*) How many teeth should a 10-inch 48-pitch spur gear have? (*b*) A gear blank measures 14″ in diameter, and is to be cut 6 pitch. How many teeth should the gear cutter be set to space?

Ans. { (*a*) 480 teeth. (*b*) 82 teeth.

(798) What is the diameter of a spur gear which has 70 teeth, and whose circular pitch is $1\frac{1}{4}$″? Ans. 27.852 inches.

(799) Given, the distance between the centers of two gears = $15\frac{1}{2}$″. What must be their diameters, so that the ratio of their speeds will be as 3 is to 2?

Ans. $18\frac{3}{5}$ and $12\frac{2}{5}$ inches.

(800) (*a*) How long is the shortest arc of action that can be allowed with a pair of gears in running contact? (*b*) In the

epicycloidal system, why is the generating circle never made larger than $\frac{5}{8}$ of the diameter of the smallest wheel of the set?

(801) What can you say for and against both the epicycloidal and involute systems of gearing?

(802) It is desired to lay out the tooth curves for an epicycloidal internal gear and pinion; diametral pitch, 3; number of teeth in gear, 30; number in the pinion, 21. What diameter-generating circle should be used? Ans. $1\frac{1}{3}$ inches.

(803) Determine whether two 4-pitch involute gears of 12 and 16 teeth, respectively, would interfere when running together, without rounding the points of the teeth. Take the angle of obliquity = 15°.

(804) Lay out two bevel-gear blanks from the following data: Shafts at right angles; ratio of speed, 4 : 3; diametral pitch, 4; number of teeth in largest gear, 24; length of face of gears, $1\frac{1}{2}''$. Write all necessary dimensions on the drawing. It is not necessary to draw the hubs.

(805) A triple-threaded worm drives a worm gear having 40 teeth. How many revolutions of the wheel will be caused by 100 revolutions of the worm? Ans. 7.5 revolutions.

(806) Given, the distance between the centers of the shafts of a worm and worm-wheel = $6''$; ratio of speeds = 40 : 1; pitch of (single-threaded) worm thread = $\frac{3}{4}''$; the thread and teeth are made according to the involute system. The length of the worm is to be $4''$, and the diameter of the worm shaft $1\frac{1}{4}''$. The teeth of the gear are to be proportioned according to Table 34, column 2.

(*a*) Calculate the number of teeth, pitch diameter, outside diameter, and whole depth of gear.

(*b*) Make a drawing showing the longitudinal section of the worm; write all needed dimensions on the drawing.

(807) A worm 2 inches in diameter, having 3 threads per inch, is to drive a spur gear having slanting teeth. What angle should the teeth make? Ans. 86° 58′, nearly.

(808) A ratchet having 72 teeth is attached to the end of a screw having $11\frac{1}{2}$ threads per inch. How much "feed" would the screw give for every ratchet tooth moved? Ans. $\frac{1}{828}$ inch.

APPLIED MECHANICS.

(ARTS. 1589–1686.)

(809) A Prony brake, with a lever arm $4\frac{1}{2}$ ft. long, was applied to a pulley running at 275 revolutions per minute. The lever was found to balance with 70 lb. in the scale pan. How much power did the brake absorb? Ans. 16.49 H. P.

(810) If you desired to find the brake horsepower of a steam engine which was rated at 6 H. P., and ran at 200 revolutions, and had on hand a number of weights aggregating 50 lb., about what length lever arm would you provide for?

(811) A rope brake made of rope of $\frac{8}{10}$ inch in diameter is used to test a gas engine. The diameter of the band wheel is 48 inches; reading of spring balance, 8 lb.; weights used, 315 lb.; number of revolutions per minute, 160. Find the horsepower. Ans. 19.01 H. P.

(812) (*a*) If a valve has neither lap nor lead, what will be the angle of advance of the eccentric? (*b*) At what point will cut-off occur? (*c*) Would such an engine run? (*d*) What effect would adding lap have on the cut-off?

(813) Name in their order the different events occurring in both ends of the cylinder (taken together) of a steam engine during one complete revolution.

(814) (*a*) How do you tell in which direction a slide-valve engine having a reversing rocker will run? (*b*) If no rocker is used, about what position will the eccentric occupy relative to the crank?

(815) What effects would increasing the lap have upon admission, cut-off, and port opening, supposing the angle of advance to remain unchanged?

(816) What alterations would have to be made on a slide-valve engine to hasten the cut-off without changing the lead?

(817) What is the effect of inside lap?

(818) In valve setting, what effect on the distribution of steam has (*a*) increasing the angle of advance? (*b*) Lengthening the eccentric rod? (*c*) Shortening the valve stem?

(819) Engines are usually set to give equal leads; but if you were called upon to set a slide-valve engine for equal cut-offs, how would you do it?

HINT.—The cross-head positions for cut-off at each end must be found, and the valve must be so adjusted that these positions will be at equal distances from the ends of the stroke.

(820) If the lead of a valve be increased, in what way will it affect the port opening?

(821) (*a*) At what position of a slide valve is its displacement the least? (*b*) Sketch a diagram similar to Fig. 444, but with the crank at position $O R_1$, and indicate what the displacement of the valve will be for the position.

(822) (*a*) Having given the lap of a valve and the displacement, how would you find the port opening for that position? (*b*) Show by a diagram, accompanied by a brief explanation, how this principle can be applied to a displacement diagram, so that the crank positions for admission, cut-off, etc., can be found.

(823) An engine is to have a stroke of 40 inches, a cylinder diameter of 18 inches, and is to run at 90 revolutions per minute. Assuming the ports to be 18 inches long; the width of the bridges, $\frac{15}{16}$ inch; inside lap, $\frac{1}{8}''$, and the travel, 6 inches, calculate (*a*) the width of the steam ports; (*b*) the maximum port opening; (*c*) the width of the exhaust port.

(824) Given, lap, $\frac{7}{8}$ inch; width of port opening, $1\frac{5}{8}$ inches; travel, 5 inches; lead, $\frac{1}{16}$ inch. Draw a diagram and indicate the angle through which the crank is moving (*a*) while the port is opening, and (*b*) while the port is closing.

(825) Given, stroke of an engine, 18 inches; length of connecting-rod, 45 inches; travel of valve, $4\frac{3}{4}$ inches; angle of advance, 35°. Cut-off is to take place at exactly $13\frac{1}{2}$ inches

on each stroke, and exhaust at exactly 17 inches. Draw a separate diagram for each end of the cylinder, showing the crank positions at which the events take place, the outside and inside laps, and the leads. Draw the valve circles half size and the crank circles to a scale of 3 inches = one foot. Care should be taken to get the different lap circles in the right places.

(826) What measurements would you take from an actual engine in order to draw a valve diagram for the engine ?

(827) Given, point of cut-off, $\frac{3}{4}$ stroke; lap, $\frac{3}{4}$ inch; lead, $\frac{1}{8}$ inch. Find travel and angle of advance by the aid of the diagram.

Ans. Travel, $3\frac{1}{4}''$; angle of advance $= 32\frac{1}{2}°$, nearly.

(828) Draw the valve diagram and a section of the valve and ports, and find the throw and position of the eccentric from the following data: Stroke, 26 inches; diameter of cylinder, 17 inches; lead, $\frac{1}{8}$ inch; cut-off, $\frac{3}{4}$ stroke; release, $\frac{59}{64}$ stroke. The engine is to run "over" at 150 revolutions per minute, and the valve is to be moved through a reverse lever, the length of the valve arm being to the length of the eccentric arm as 5 : 4. Assume width of bridge $= 1''$.

(829) (*a*) Why are piston valves used ? (*b*) Can you think of any possible disadvantage that they might have ?

(830) What change in the position of the eccentric is necessary if the valve is to take steam inside ?

(831) A Trick valve is to be designed. Given, width *t*, in Fig. 457 $= \frac{5}{8}''$; width of port $S = 1\frac{1}{2}''$; inside lap, $\frac{1}{8}''$; width of bridge, $\frac{7}{8}''$. Required (*a*) width of passage through valve; (*b*) valve travel; (*c*) width of cylinder face *c*; (*d*) width of exhaust port.

Ans. $\begin{cases} (a) & \frac{7}{16}''. \\ (b) & 3\frac{1}{4}''. \\ (c) & 1\frac{1}{2}''. \\ (d) & 2\frac{1}{4}''. \end{cases}$

(832) What is to be gained by using a double-ported valve ?

(833) The lengths of the crank and connecting-rod of a certain engine are in the ratio of 1 to 4. If the cut-off on

the return stroke occurs at $\frac{2}{3}$ stroke, at what point does cut-off take place on the forward stroke, the lead remaining constant? The travel of the valve is 3 inches.

Ans. $\frac{37}{48}$ stroke.

(834) What advantage is to be gained by using a Meyer valve?

(835) (*a*) In the Meyer valve gear, which valve cuts off and which admits the steam? (*b*) Which gives the compression and exhaust?

(836) State the principal advantage to be obtained from a shaft governor.

(837) With a double-valve gear.

(838) A Prony brake is used to test a small water-wheel. If the length of the arm is 40 inches, weight in scale pan 2 lb. 10 oz., and revolutions per minute 263, what is the brake horsepower of the wheel? Ans. .438158 H. P.

(839) Make a sketch showing the relative positions of the crank and two eccentrics in the Stephenson link motion, (*a*) when no rocker is used, and (*b*) when a reversing rocker is used.

(840) How can you tell when a link motion has "open rods"?

(841) The following questions relate to a link motion with "open rods":

In mid-gear, (*a*) what is the greatest possible displacement of the valve equal to? (*b*) What is the greatest travel equal to? (*c*) The greatest port opening? (*d*) What is the travel equal to in full gear? (*e*) In passing from mid-gear to full gear, how do the lead and cut-off vary?

(842) (*a*) In what ways is the effect of a link with "crossed rods" different from one with "open rods"? (*b*) If you wished to design an engine to stop when the link was placed in mid-gear, would you use "open" or "crossed" rods?

(843) When the link motion is to be used as a variable cut-off gear, why should it be designed to make the valve over-travel in full gear?

(844) (*a*) What is the object in curving the links of a Stephenson link motion? (*b*) Would it answer to make them straight, in case the motion was to be used on a hoisting engine, and why?

(845) (*a*) What is meant by slip? (*b*) Suppose a link motion were to be used on a marine engine which ran almost entirely in one direction. What manner of suspension of the link would cause the least wear of the link?

(846) What is to be gained by using two eccentrics on a Corliss engine, one for operating the steam and the other for operating the exhaust valves?

(847) (*a*) Why must the steam valves of a Corliss engine have lap? (*b*) What is the disadvantage in using lap on these valves?

(848) (*a*) Why must the exhaust valves of a Corliss engine have lap? (*b*) Could not the same effect be produced by shortening the exhaust rods, thus turning the valve away from the edge of the port, so as to give it lead opening? Give reasons.

(849) Explain how a rapid movement of the steam valves is secured in a Corliss engine.

(850) Is it feasible to regulate the cut-off by changing the angle of advance only? Why?

(851) An engine with 24″ stroke is to have a range of cut-off varying from $\frac{1}{12}$ to $\frac{5}{6}$ stroke, the lead remaining constant at $\frac{1}{8}$ inch. Port opening for cut-off at $\frac{5}{6}$ stroke = 1″.

(*a*) Determine, by drawing a diagram, what must be the movement of the eccentric across the shaft to produce this result. Draw the crank circle $\frac{1}{4}$ size and the valve diagram full size. (*b*) What is the port opening for cut-off at $\frac{1}{12}$ stroke?

(852) A slide valve is to be operated by a Stephenson's link motion. The point of cut-off is to be .8 stroke; full gear lead, $\frac{1}{16}$″; width of steam ports, $1\frac{1}{4}$″; width of bridge, $\frac{7}{8}$″; inside lap, $\frac{1}{8}$″; over-travel in full gear, $\frac{3}{8}$″. No rocker is to be used, the valve being in line with the center line of the

link motion. The distance from the center of the shaft to the middle position of the center of the link arc is 3 ft.

Draw skeleton diagrams of the link in full gear and in mid-gear, with a section of the valve and ports. The latter may be placed above the link, as in Fig. 467. The valve diagram should be drawn $\frac{1}{2}$ size, and the ports and link diagram $\frac{1}{4}$ size. Assume the length of the link to be 12″.

Also state what dimensions you get for the following: Outside lap; full gear travel; angle of advance; mid-gear lead.

(853) What is meant (*a*) by the "height" of a governor, and (*b*) by the "sensitiveness" of a governor?

(854) The height of a simple revolving pendulum is 20″. (*a*) How many revolutions per minute must its speed be increased to make the height 18″? (*b*) How much must the speed be decreased to make the height 22″?

Ans. $\begin{cases} (a)\ 2.272 \text{ revolutions.} \\ (b)\ 1.955 \text{ revolutions.} \end{cases}$

(855) (*a*) What objections are there to the ordinary pendulum governor? (*b*) The height of a single pendulum governor running at a certain speed is 1 inch; what would be the height of a weighted governor running at the same speed, supposing the weight to be five times as heavy as both balls taken together? Ans. 6 inches.

(856) If an engine were fitted with a weighted pendulum governor, how could you increase its speed?

(857) In designing a weighted governor, the following dimensions and weights were obtained from the drawing: (Refer to Fig. 481.)

$$x = 6.5; \qquad r = 9;$$
$$H = 10; \qquad n = 5.4;$$
$$y = 14; \qquad D = 4;$$
$$m = 5.85; \qquad E = 3.$$

Assuming $N = 240$ rev. per min. and $R = 5$ lb., what should be the weight of each ball and of the counterpoise?

(858) Under what conditions would you use a transmission dynamometer?

STEAM BOILERS.

(ARTS. 1687–1900.)

(859) (*a*) What is meant by *internally fired boiler?* (*b*) *Externally fired boiler?*

(860) Enumerate the various boiler mountings, and state the duty of each.

(861) Find the necessary thickness of a boiler tube which is 3″ in diameter, 15 ft. long, and is subjected to an external steam pressure of 80 lb. per sq. in. Ans. $\frac{3}{16}$″, nearly.

(862) Find the diameter of the welded wrought-iron stay-rods of a Scotch boiler carrying a pressure of 125 pounds, the rods being pitched 12 inches apart one way and $13\frac{1}{2}$ inches the other. Ans. $2\frac{1}{16}$″, nearly.

(863) What should be the thickness of the walls of an iron combustion chamber, stayed by iron staybolts with nuts, pitched 8 inches apart, the steam pressure being 150 lb.? Ans. $\frac{1}{2}$″.

(864) The composition of a fuel is: Carbon, 80%; hydrogen, 8%; oxygen, 6%; ash, 6%. (*a*) Find the heat of combustion per pound of fuel. (*b*) Find the minimum air supply per pound of fuel. Ans. $\begin{cases} (a)\ 16{,}097.32 \text{ B. T. U.} \\ (b)\ 11.8 \text{ lb.} \end{cases}$

(865) If the temperature of the furnace fire is 2,800° F., what is gained in economy by reducing the temperature of the flue gases from 600° to 250° F.? Ans. 12.5%.

(866) A boiler evaporates 28,930 pounds of feed-water from a temperature of 127° into steam at 85 pounds pressure during a run of 10 hours. What horsepower is developed by the boiler? Ans. 94.3 H. P.

(867) In a test of the quality of steam made by a barrel calorimeter, the weight of cold water was 360 pounds and

its temperature 44° F. After steam was run in, the total weight of water in the barrel was 382 pounds, and the temperature had risen to 110°. The steam pressure being 80 pounds, what was the quality of the steam? Ans. 97.7%.

(868) Allowing a maximum velocity of flow of 4,000 feet per minute, what should be the minimum diameter of a steam pipe to carry 7,200 pounds of steam per hour at a pressure of 75 pounds? Ans. $5\frac{3}{16}''$.

(869) A safety valve has a diameter of opening of 4 inches. The weight is 107 pounds. The distance from the valve to fulcrum is $4\frac{1}{2}''$; distance from fulcrum to center of gravity of lever, 19″. Weight of valve and stem, $8\frac{1}{2}$ lb.; weight of lever, 21 lb. Graduate the lever for the pressures 40, 45, 50, 55, and 60 pounds per sq. in.; that is, give distances from fulcrum for these pressures.

(870) (*a*) What are the relative advantages and disadvantages of the plain cylindrical type? (*b*) In what localities may it be economically used?

(871) A cylindrical boiler shell is 12 feet long, 4 feet 3 inches in diameter, and subjected to a steam pressure of 80 lb. per sq. in. (*a*) What is the total force tending to rupture the shell along *one* longitudinal seam? (*b*) What is the total force tending to rupture it along a girth seam?

Ans. $\begin{cases} (a)\ 293{,}760 \text{ lb.} \\ (b)\ 163{,}426 \text{ lb.} \end{cases}$

(872) (*a*) What is a lap joint? (*b*) A butt joint? (*c*) Distinguish between single and double-riveted joints; (*d*) between chain riveting and staggered riveting. (*e*) Which is the strongest of the ordinary riveted joints, and why?

(873) Steel stayrods $1\frac{3}{4}$ inches in diameter support the heads of a boiler under a pressure of 140 pounds. Supposing them to be pitched equidistant horizontally and vertically, how far apart may they be placed?

Ans. $12\frac{1}{2}$ in., nearly.

(874) (*a*) Describe a boiler-feed apparatus. (*b*) In what part of the boiler should the feed be introduced, and why? (*c*) Describe a feed-water heater; (*d*) an economizer.

(875) (*a*) What fuels are chiefly used for steam making? (*b*) Describe the different varieties of coal. (*c*) 400 pounds of wood is equivalent to how many pounds of coal? (*d*) What are the advantages and disadvantages of liquid and gaseous fuels?

(876) The average temperature of the external air being 60° F., and the temperature of the flue gases 580° F., what must be the height of the chimney to produce a draft of .77 inch of water? Ans. 110 feet.

(877) State fully the various points to be considered in the arrangement of the tubes in a return tubular boiler.

(878) In a test of the quality of steam, two separator calorimeters were coupled together; 14 ounces of water were collected in the first separator, and 2 ounces in the second. The amount of dry steam condensed was $20\frac{1}{2}$ pounds. Find the quality of the steam. Ans. $96\frac{1}{2}$%.

(879) Describe a separator; explain the principle of its action.

(880) (*a*) Describe the Cornish, Lancashire, and Galloway types of boilers. (*b*) Where are they used chiefly? (*c*) What types of boilers are mostly used in the United States?

(881) A cylindrical boiler shell is 18 ft. long, 4 ft. 6 in. in diameter, and subjected to a pressure of 65 lb. per sq. in. (*a*) What is the force tending to rupture *one* longitudinal seam *per inch of length?* (*b*) What is the force *per inch of length* tending to rupture a girth seam? Ans. $\begin{cases} (a)\ 1{,}755 \text{ lb.} \\ (b)\ 877\frac{1}{2} \text{ lb.} \end{cases}$

(882) In what ways may a riveted joint fracture?

(883) What should be the diameter of copper staybolts, pitched $3\frac{3}{4}$ inches apart each way, the steam pressure being 128 pounds? Ans. $\frac{3}{4}''$

(884) If a boiler generates steam at 85 pounds pressure, what per cent. may be gained by heating the feed, originally at 65° F., by means of a feed-water heater, to 200° F.?
Ans. 11.8%.

(885) A fuel has the following composition: Carbon, 94%; hydrogen, 1%; oxygen, 2%; ash, etc., 3%. How many pounds of water should a pound of this fuel evaporate from and at 212°? Ans. 14.59 lb.

(886) What draft pressure will be given by a chimney 135 feet high, the temperature of the air being 65° F., and of the gases 420° F.? Ans. $\frac{3}{4}''$.

(887) What should be the combined area of the tubes of a return tubular boiler? What should be the space between the bridge and shell?

(888) What is incrustation? What are the common scale-forming substances? In what ways may incrustation be prevented?

(889) Describe the injector; explain its action. Why is it economical to use a much larger injector than is actually necessary to do the work?

(890) Trace the evolution of the modern water-tube boiler from the plain cylindrical boiler. Explain clearly why the water-tube type is free from danger of explosion.

(891) A boiler shell is 38 inches in diameter, and withstands a steam pressure of 110 lb. per sq. in. Assuming the safe working strength of the iron plate to be 9,500 lb., and the efficiency of the joint to be 60 per cent., what should be the thickness of the plate? Ans. $\frac{3}{8}''$.

(892) Assuming the following data, let the student design the riveting of a steel boiler shell 5 feet in diameter: $S_1 = 63,000$ lb.; $S_3 = 56,000$ lb.; $S_2 = 100,000$ lb.; $f = 5$, and steam pressure, 120 lb. Design a double-riveted lap joint, using formulas (*a*), (*b*), (*c*), (*d*), and Table 37 as a guide for the proportions.

(893) What should be the diameter of steel screw staybolts spaced $4\frac{1}{2}$ inches apart each way, the steam pressure being 160 pounds? Ans. $\frac{3}{4}''$, nearly.

(894) (*a*) A boiler of 120 horsepower, using 30 pounds of water per horsepower per hour, and generating steam at a pressure of 70 pounds per sq. in., requires a safety valve

of what area? (*b*) Supposing the valve to be of the dead-weight type, what should be its weight?

Ans. { (*a*) $22\frac{1}{2}$ sq. in. (*b*) 1,575 lb.

(895) In what way does too great a supply of air to the fire lessen the economy of the combustion? Why does too small a supply of air result also in loss? Explain fully.

(896) (*a*) Calculate the diameter of a round chimney 105 feet high, furnishing draft for 700 horsepower of boilers. (*b*) If square, what should be the length of a side?

Ans. { (*a*) 65.3″. (*b*) 58.3″.

(897) A return tubular boiler of 150 horsepower is expected to evaporate $10\frac{1}{4}$ pounds of water per pound of coal from and at 212°. The draft will allow a rate of combustion of 16 pounds of coal per square foot of grate per hour. If anthracite coal is to be used, what heating surface would you give the boiler?

(898) Describe the various forms of internal corrosion, and give their causes. What precautions must be taken to prevent both internal and external corrosion?

(899) What is the maximum amount of bituminous coal that can be burned in 10 hours on a grate $3\frac{1}{2}$ ft. × 5 ft., with a chimney 135 feet high? Ans. 4,399.5 lb.

(900) (*a*) What type of boiler is chiefly used at present for marine service? Describe it. (*b*) What are the points for and against the use of water-tube boilers for marine work?

(901) A steel marine boiler is 13 feet in diameter, and works under a steam pressure of 140 lb. per sq. in. Supposing the tensile strength of the material to be 64,000 lb., the efficiency of the joints 72 per cent., and assuming a factor of safety of 4, what should be the thickness of the plates? Ans. $\frac{15}{16}$″, nearly.

(902) Find by calculation (*a*) the proper pitch, and (*b*) the diameter of the rivet holes for a double-riveted lap

joint, the plates being $\frac{3}{8}$ inch thick. (*c*) What is the efficiency of the joint?

Ans. $\begin{cases} (a)\ 3\frac{3}{8}''. \\ (b)\ \frac{3}{4}''. \\ (c)\ 78\%. \end{cases}$

(903) Wrought-iron screw staybolts are 1 inch in diameter, and resist a pressure of 150 pounds per sq. in. If spaced equally distant each way, how far apart should they be spaced? Ans. 5.6″.

(904) (*a*) Describe the different types of safety valves. (*b*) What precaution must be observed in attaching the safety valve to the boiler? (*c*) What care do safety valves require?

(905) A grate is 4 feet wide by 6 feet long. The rate of combustion is 12 pounds of coal per square foot of grate per hour. Assuming the coal to be anthracite, consisting of 95 per cent. carbon, and the remainder ash, and that $1\frac{1}{2}$ times the theoretical amount of air is supplied, how many cubic feet of air at 62° F. must be furnished per minute?

Ans. 1,042.6 cu. ft.

(906) (*a*) A boiler plant of what horsepower may be supplied with draft by a chimney 125 feet high and 5 feet in diameter? (*b*) By a chimney 80 feet high and 3 ft. 8 in. square?

Ans. $\begin{cases} (a)\ 632 \text{ H. P.} \\ (b)\ 336 \text{ H. P.} \end{cases}$

(907) If a boiler generates steam at 105 lb. pressure, and receives the feed water at 120° F., what economy will result from raising the feed temperature by using an economizer to 200° F.? Ans. 7.3%.

(908) Describe the methods of making a hydrostatic test of a boiler. What are the points in favor of and against the hydrostatic test compared with the hammer test?

(909) What size of direct-acting steam pump would you adopt to feed a boiler plant of 300 horsepower?

(910) Describe the construction and operation of a Field tube.

(911) (*a*) What are the relative strengths of the longitudinal and girth seams in example 901? (*b*) Give statement and proof.

(912) (*a*) How should the joints be arranged in the furnace plates of externally fired boilers? (*b*) In which direction should the fibers of wrought-iron plates extend? (*c*) In lap joints, which way should the inside lap face, and why?

(913) Find the effective diameter of a diagonal crow-foot brace supporting an area of 35 sq. in., the steam pressure being 80 pounds. The brace is of wrought iron, welded, and makes an angle of 18° with the shell. Ans. $\frac{13}{16}''$.

(914) Given the following data concerning a lever safety valve: Diameter of valve opening, 5″; length of straight lever, 40″; length from valve to fulcrum, 4″; weight of valve and stem, 8 lb.; weight of lever, 12 lb.; blow-off pressure, 110 lb., what should be the weight placed at the end of the lever arm? Ans. 209.2 lb.

(915) How is the transfer of heat from the furnace to the water in the boiler accomplished? For what reasons should a boiler have a good water circulation?

(916) A certain boiler plant has a chimney 90 feet high. If the plant should be fitted with economizers which reduce the temperature of the furnace gases from 620° to 270°, what additional height should be given the chimney in order that the draft shall remain unchanged? Assume the air temperature to be 60° F. Ans. 83.2 ft.

(917) What is the factor of evaporation? Compute the factor of evaporation for the following cases:

Pressure of Steam.	Temperature of Feed.	
(*a*) 87-lb. gauge.	113° F.	Ans. 1.1399.
(*b*) 117-lb. gauge.	73° F.	Ans. 1.1874.
(*c*) 134-lb. gauge.	87° F.	Ans. 1.1758.
(*d*) 38-lb. gauge.	158° F.	Ans. 1.0792.

(918) What are the causes of boiler explosions? A tank filled with carbonic acid gas at a pressure of 200 or 300

pounds may rupture, but will not explode, while the same tank filled with steam and hot water may explode with great violence. Explain.

(919) Describe the steam loop, and explain its action. What is the object of removing the water from the steam pipe and the separator?

(920) (*a*) What are the materials chiefly used in boiler construction? (*b*) What class of steel is used for plates? (*c*) What qualities should iron or steel possess if intended for use in boiler plates? (*d*) Why should not steel plates have a greater tensile strength than 65,000 pounds per square inch?

(921) (*a*) What forms of vessels are self-supporting? (*b*) Why are they self-supporting? (*c*) Which is stronger under pressure, a spherical or a cylindrical vessel of the same diameter?

(922) (*a*) What are the arguments for and against the punching of rivet holes? (*b*) What is the general practice of good makers in this connection? (*c*) What caution must be observed in the calking of riveted seams?

(923) The girder stays of a fire-box are 33 inches long, and spaced 8 inches apart; the steam pressure being 120 pounds, find the depth of the girder. Ans. $5\frac{5}{8}''$, nearly.

(924) Given the same data as in example 914, at what distance from the fulcrum should a weight of 265 pounds be placed? Ans. 31.6 in.

(925) In what ways may the heat generated in the furnace be wasted or lost? What is smoke, and what is necessary for its prevention?

(926) (*a*) With soft coal of good quality, what is the maximum rate of combustion with a chimney 80 feet high? (*b*) With a chimney 115 feet high?

Ans. $\begin{cases} (a)\ 20\frac{1}{8}\text{ lb.} \\ (b)\ 24\frac{1}{8}\text{ lb.} \end{cases}$

(927) For what purposes are boiler trials made? Describe briefly the steps to be taken in making a trial.

(928) What course should be pursued when the water is found to be low in the boiler? What should be done when the boiler foams or primes badly? How may unused boilers be kept in good condition?

(929) In a test with a barrel calorimeter, the weight of cold water is 327 pounds; the weight of steam condensed, 24 pounds; the initial and final temperature of the water, 40° and 120°, and the steam pressure, 65 pounds. (*a*) What is the quality of the steam? (*b*) What is the number of degrees of superheat? Ans. (*b*) 6.5°.

(930) (*a*) By what process is boiler steel made? (*b*) What is the objection to cast iron as a boiler material? (*c*) Describe a boiler made of cast iron.

(931) What pressure may be safely borne by a steel boiler shell 3 ft. 6 in. in diameter, the plate being $\frac{5}{16}$ in. thick; the safe working stress of the material, 12,000 lb., and the efficiency of the joint, 66 per cent.?

Ans. 118 lb. per sq. in., nearly.

(932) What pressure might be allowed on the wrought-iron head of a cylindrical boiler which is $\frac{1}{2}''$ in. thick and 48″ in diameter, the safe working stress of the iron being 10,000 pounds, *if the head is unstayed?*

Ans. 6.5 lb. per sq. in.

(933) What pressure may be resisted by girder stays 42 inches long, $7\frac{1}{2}$ inches deep, and spaced 10 inches apart?

Ans. 143.5 lb. per sq. in.

(934) (*a*) Describe the working of a Bourdon pressure gauge. (*b*) Describe the apparatus used to obtain the water-level. (*c*) What is a fusible plug, and what is its object? (*d*) What are the relative advantages of domes and dry pipes? (*e*) What is the purpose of manholes and mudholes?

(935) Describe the different methods of obtaining forced drafts. For what fuels are the forced draft and the shaking grate particularly adapted? How is the forced draft of the locomotive obtained?

(936) Under ordinary conditions, what is the minimum height of a chimney that will permit a rate of combustion of anthracite coal of 14 pounds per sq. ft. per hour ?

Ans. 100 ft.

(937) (*a*) What is the efficiency of a boiler ? (*b*) A boiler plant during a trial uses 19,000 pounds of coal, of which 93% is combustible, and evaporates 175,000 pounds of water into steam at 60 pounds pressure from a feed temperature of 52° F. If the heat of combustion of a pound of the coal is 13,500 B. T. U., what is the efficiency of the plant ?

Ans. (*b*) 78.8%.

(938) Explain the action of zinc in preventing corrosion and incrustation.

(939) Upon what principle does the feed-water purifier depend ?

(940) (*a*) Describe the construction and operation of the Heine boiler. (*b*) What, if any, is the advantage of suspending the mud drum in the boiling water ?

(941) In a calorimeter test, the quality of the steam was found to be 101.48%. The gauge pressure being 83 lb., how much was the steam superheated ? Ans. 27.46°.

(942) What would be the necessary thickness of the cast-iron head of a cylindrical boiler 2 ft. 10 in. in diameter, the steam pressure being 75 pounds, and the allowable safe stress in the cast iron being 5,000 pounds ? Ans. 1.7″.

(943) What safe pressure may be allowed on a steel plate $\frac{5}{16}$ inch thick, stayed with steel screw staybolts spaced $4\frac{1}{2}$ inches apart each way ? Ans. 135 lb.

(944) How many B. T. U. are obtained by the combustion of a pound of carbon with sufficient air to form (*a*) carbon dioxide, (*b*) carbon monoxide ? (*c*) Distinguish between complete and incomplete combustion.

(945) Required the power necessary to furnish a forced draft for a grate area of 175 sq. ft., with a rate of combustion of 40 lb. per sq. ft. per hour. The pressure of the blast is 12 lb. per sq. ft. ; the volume of air required is 230 cu. ft.

per pound of fuel per minute, and the efficiency of the apparatus is 62.5 per cent. Ans. 15.61 H. P.

(946) A certain type of boiler is expected to evaporate $9\frac{1}{2}$ pounds of water from and at 212° per hour per pound of coal. What grate surface must be given the boiler if it is to generate 3,000 pounds of steam per hour at 212°, the desired rate of combustion being 15 pounds per sq. ft. per hour? Ans. 21 sq. ft.

(947) Taking the data of question 937, find (*a*) the equivalent evaporation from and at 212° F.; (*b*) the evaporation from and at 212° per pound of coal; (*c*) per pound of combustible; (*d*) the horsepower developed by the plant, the duration of the test being 22 hours.

Ans. (*a*) 209,331 lb.; (*b*) 11.02 lb.; (*c*) 11.85 lb.; (*d*) 275.8 H. P.

(948) What are the objects of a boiler setting? What precautions must be observed in hanging or supporting very long boilers? Describe the setting of the return tubular boiler; of the Lancashire and Galloway boilers.

(949) What quantity of water can theoretically be evaporated from a temperature of 100° into steam of a temperature of 240° by a pound of lignite, the chemical composition of which is carbon, .643; hydrogen, .042; oxygen, .100; miscellaneous, .215? Ans. 10.26 lb.

(950) What is the horsepower of a boiler which generates 935 pounds of steam per hour at a pressure of 40 pounds, taking the feed at 55° F.? Ans. 32.2 H. P.

(951) Suppose the material of a furnace flue which is $\frac{3}{8}''$ thick and 30″ in diameter to have a safe working tenacity of 12,000 lb. (*a*) What *internal* pressure would it safely support? (*b*) Why will a cylindrical vessel safely stand a greater internal than external pressure?

Ans. (*a*) 300 lb. per sq. in.

(952) What would be the necessary thickness of an unstayed steel flat plate 20 in. square, subjected to a pressure

of 30 lb. per sq. in., the allowable safe stress in the steel being 12,000 lb.? Ans. $\frac{1}{2}''$.

(953) What should be the thickness of an iron fire-box plate subjected to a pressure of 110 pounds per square inch and stayed by iron screw stays pitched 4 inches apart?

Ans. $\frac{9}{32}''$, nearly.

(954) If 18 pounds of air are furnished to each pound of the fuel of example 864, what will be the temperature of the furnace? Ans. 3,297°.

(955) Describe the different systems of hand firing. What are the advantages of the mechanical stoker? To what fuels is the mechanical stoker particularly suited?

(956) If in example 866, 3,120 pounds of coal were burned during the run, what was the evaporation from and at 212° per pound of coal? Ans. 10.43 pounds.

(957) What is meant by the "quality of the steam"? Explain the operation of the barrel calorimeter, and clearly state the principle on which its action depends.

(958) In what ways may the main piping of a boiler be arranged to allow for expansion? If a copper bend is placed in a steam log carrying a pressure of 105 pounds, what must be the thickness of the copper pipe, the diameter being $4\frac{1}{2}$ inches? Ans. .172″, say $\frac{11}{64}''$.

INDEX.

T. PAGE